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MULTIPLE-APPLICATION 100 METER LUNAR DRILL  
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A TRANSPORTABLE, MULTIPLE-APPLICATION  
100 METER LUNAR DRILL

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Mr. J.W. Brazell

Design Group #2:

Rodney L. Case

Buckley M. Collum

Clifford S. Halstead

Michael D. Harper

Peter G. Stangel

Mark E. Zimmerman

# ABSTRACT

The purpose of this report is present a design of a semi-transportable, multiple-application lunar drill. The paper focuses on the design of the drill rig, drive system, anchoring system, drill stem, and cooling system. The full effects of the lunar environment were accounted for in the design.

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## INTRODUCTION

Interest in the moon dates back to the dawn of mankind. In the past two decades, man has learned much about the solar system in which he exists, due to the advent of space travel. Though man has landed on the moon, he knows very little of its substrata.

Our group set out to design a drill which would adequately serve in the lunar environment. We wanted to make it light and semi-portable, yet strong and durable. We wanted it to be easily usable by a technician encumbered by his lunar-environment-suit. We designed it to have a good service life since spare parts would have to come from Earth.

Our result is a tripodal drill rig using two 240 volt DC motors: one for rotary action and one for vertical action. It uses an auger system for anchoring. The drill string is composed of up to 27 sections, with two auger flight sections for particle removal. The bit is a self-sharpening diamond matrix bit designed for constant velocity, variable pressure applications.

Heat created by friction and electrical motor activity is dissipated using radiation. Hydrogen is forced over the components to collect the heat. The excess heat is then radiated to deep space.

## BACKGROUND

In the Apollo 16 and 17 missions, two drills were tested; one hand-operated and one electrically-powered, respectively. However, due to the limitations of the space-craft cargo area, these drills were small and were developed for depths under 10 m. These were also auger-type drills, so samples obtained could not be specified for a given depth.

These samples offered researchers much information on the lunar surface, but still left many questions about the deeper substrata of the moon unanswered. The knowledge gained from this information would reveal the moon's composition, thermal characteristics, and history , which might give insight to how the moon was formed.

A drill had to be developed which would be able to take core samples from deep in the sub-surface of the moon. These core samples would yield the information desired by researchers.

The drill could also be used for additional applications on the moon. It could be used to install anchoring systems for structures, buildings, or mining supports. It could also serve to bore holes for pipes and lines to mine shafts or sub-surface complexes.

## DRILL RIG DESIGN

Prior to choosing the present dual drive, chain-cable rig, various designs were studied. The designs were generally ruled out because they were too impractical or too complex.

A design involving a power screw method of vertical driving was studied. This design would assure our requirement of a constant feed of 2.0 in/min, but would undergo endurance or fatigue failure due to the large number of cycles necessary to drive the 23 - 4 meter sections. Power screws are not designed to undergo power loading at low speeds and subsequent high speed retraction, over a long life of many cycles.

This was followed up with a hydraulic system to control the vertical drive and retraction of the drill rod. This system offered both the advantages of a slow, high-pressure feed and quick retraction. However, it was felt that due to the necessary sealing of an inordinant number of seal and joint areas by something other than rubber (petroleum based) seals, but still retaining the necessary sealing properties, it is still beyond any technology to which this group had access. Also, in the event of a rupture of the system, the subsequent loss of all the hydraulic fluid to the near-zero pressure environment (a very quick event due to the high pressure system of the hydraulic system), and the contamination of the internal environment of the hydraulic system, due to the fluid

turning to "dust," would totally destroy the entire vertical drive system of the drill. The leak would have to be repaired and the system purged and filled in the lunar environment; we could not find any articles dealing with near zero atmosphere welding, nor could we think of a method to repurge the system in the near-zero pressure environment.

Belt drives for either the vertical drive or the rotary drive were eliminated due to the fact that no belt materials could meet the stringent conditions: too high ( $+150^{\circ}\text{C}$ ) and too low ( $-150^{\circ}\text{C}$ ) a temperature range, dust too abrasive resulting in faster wears, and friction of the belt to a driving rod very high resulting in an extreme heat which could not be convected off.

Traction drive was studied, but was found impractical for this situation, due to the complexity and quantity of the linkages necessary to meet our conditions of the vertical and rotary driver, yet maintain the low heat generation, from friction, which would serve to damage the drill rig.

Our final determination was to go with an indirect drive from the base of the tripod legs for the vertical drive, while utilizing a second motor for the rotary action of the drill rod.

Calculations and specifications for drill rig frame components are located in Appendix A.

## DRIVE MOTORS

The rotary drive motor is a vertically oriented motor mounted above the shaft. Therefore, a direct drive system is established with only a speed control/tachometer unit and a gear reduction unit between the motor and the drill shaft.

The motor is a 10 hp DC motor supplying 4.90 hp at the bit surface. It runs on 240 dc volts. The motor over its drilling life is to run at a constant rate of 210 RPM when boring. The motor chosen has a range of 300 to 900 RPM.

Attached to the motor is a disc-type braking system delivering an effective torque of 175 ft-lbs. This is to be used primarily in the event of an emergency, such as shaft lock-up, bit damage, or possible injury to personnel.

A tachometer generator is also attached to the motor. Located on the commutator end, the tachometer generator serves as a speed indicator and a speed regulator. The one chosen is a high-accuracy device to insure that the desired rate of rotation is not exceeded, which could result in bit failure.

The entire drive and control unit is hermetically sealed, along with a cooling jacket, to insure that the lubrication used in the motor does not lose its lubricating properties. and to insure that dust does not enter and damage the system. The cooling jacket is used to maintain an ambient temperature for the motor to work, in which it

is not beyond the acceptable limits specified by the builder ( $0^{\circ}\text{C}$  -  $40^{\circ}\text{C}$ ).

The downward drive is to be a constant 2 inches per minute (0.00952 inches per revolution of the drill rod). This is to be maintained by a 10 hp DC motor located under the tripod, behind the shaft control assembly. Here, it serves to control the indirect drive on the primary pair of tripod legs.

The drive motor turns a pinion at a constant rate of speed. This is connected to the indirect drive on the legs by a roller chain. The chain turns a sprocket at a constant angular velocity, thereby turning 4 spindels per side. These spindels, when driven, retract the  $1/8"$  - 7 x 19 airshaft cable. The cable is woven through sheaves, which are located on the bottom of the rotary drive assembly, to a complementary drum on the opposite leg. As the indirect drive shaft is rotated, the spindels take up the wire rope, which pulls the shaft down.

For details concerning the specifications of the drive motor systems, see Appendix B.

## LUBRICATION FOR LUNAR DRILL RIG

### Introduction

Lubrication is an integral factor in the length of the working life for a mechanical system. In a situation where two parts are in moving contact, a shear force is found to develop. This shear is directly related to the friction between the two surfaces. If the coefficient of friction can be reduced, then the wear life of the components is increased.

In the drill scheme, rotation and translation between parts can be found in a vast number of places. At these places, a means of lowering the coefficient of friction must be found or wearing and failure will develop. In a device as integrally connected and dependent on the total functioning of the whole system as the drill rig is, this could mean hampered results, total disfunction, or injury to additional equipment or personnel.

Lubrication is necessary to maintain the proper condition of these working parts. However, on the moon, normal types of lubrication are useless. The best alternative is a solid lubricant, based on molybdenum disulfide.

## Lubricant for Drill Rig

The moon's environment makes the use of normal lubricating fluids impossible. The main reasons are the temperature range and the near-zero atmospheric pressure.

The temperature range of the moon offers too large of an extreme for most liquid lubrication. Also, the near-zero pressure causes the fluid lubrication to turn to dust. Although the lubricating additives might remain after their exposure to this lack of pressure, they fail to retain the necessary cohesive bond with the materials they are supposed to protect.

Solid lubricants seem to meet the requirements necessary to protect the parts where a liquid lubricant would fail. The solid lubricant will not drastically change under a low-pressure application. It also offers good protection over a large established region.

We sought to find a substance which would handle a temperature range from  $-100^{\circ}\text{C}$  to  $+200^{\circ}\text{C}$ . It also had to be compatible with both aluminum alloys and steel for the aluminum/aluminum, aluminum/steel, and steel/steel interfaces present on the drill rig.

Molybdenum disulfide ( $\text{MoS}_2$ ) based lubricants seemed to be the best possible lubricant for this application. As with most solid lubricants, it has certain characteristic advantages:

1. Good stability at extreme temperatures.
2. Improved dynamic mechanical stability; bearings can

be placed closer to heat sources allowing the use of shorter rotating shafts, thereby alleviating problems of shaft critical speed.

3. Can be used under high nuclear radiation conditions.
4. Can be used under extremely high-load conditions.

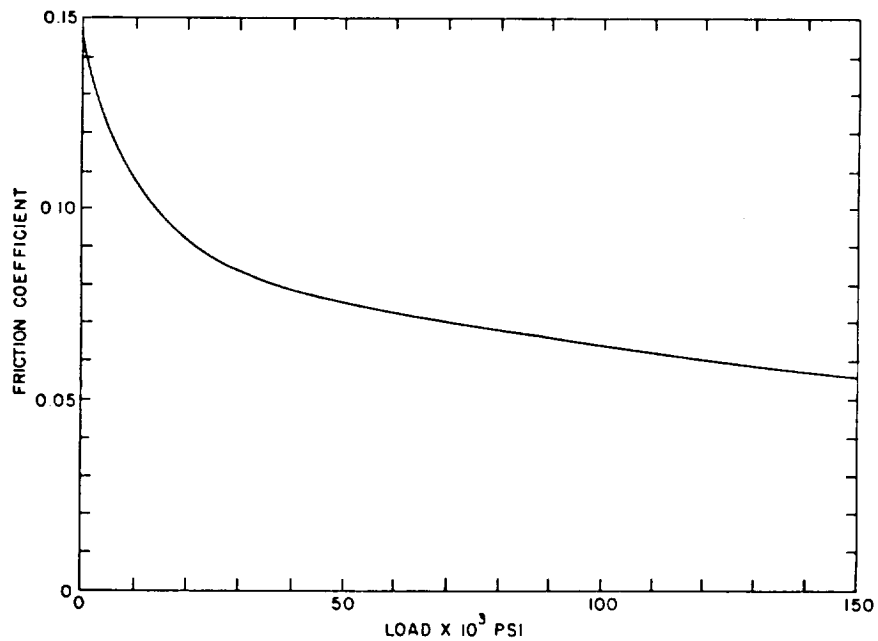
However:

1. Coefficient of friction is generally higher than with hydrodynamic lubrication.
2. Some wear is unavoidable, because of solid sliding contact.
3. Film coatings have finite wear lives.
4. Solid lubricants have no cooling capacity.

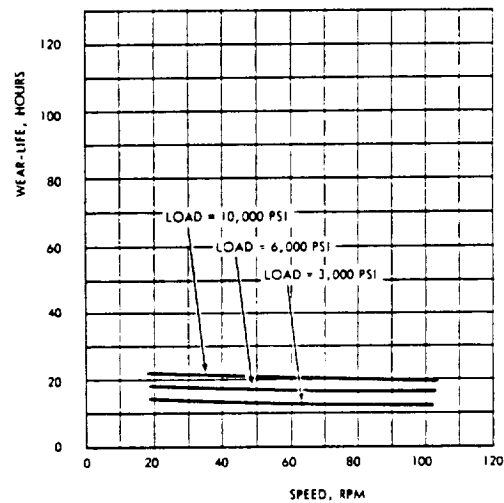
The inorganic bonded films ( $\text{MoS}_2$ , graphite, sodium silicate) are designed for a wider temperature range than are the resin-bonded materials. They are usable from  $-300^\circ\text{F}$  to over  $1200^\circ\text{F}$  in a vacuum. However, on the upper end of the scale,  $\text{MoS}_2$  may oxidize to become  $\text{MoS}_3$ . Fortunately, this weakness will not effect our consideration, since our highest temperature is under  $500^\circ\text{F}$ . Also, there is virtually no atmosphere on the moon which causes oxidation of the lubricant.

The  $\text{MoS}_2$  based films also have a reduced friction coefficient as the load is increased. This is shown in the figure on the following page. Also shown is the wear life in hours versus speed, and the wear life in hours versus load. These offer proof that the increasing usefulness of  $\text{MoS}_2$  as an acceptable pigment in inorganic resin-bonded films.

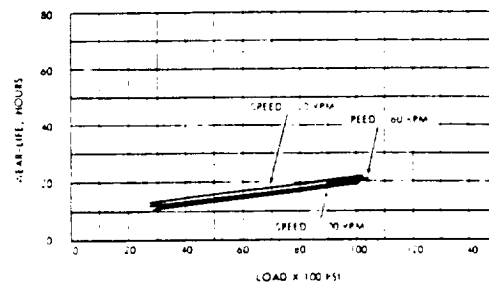
Plot of Friction Coefficient vs. Load  
for Molybdenum Disulfide Based Films



Speed vs. Wear Life of Polymeric Bonded Film



Load vs. Wear Life of a Polymeric Bonded Film



## Conclusion

A resin-bonded film with a pigment of molybdenum disulfide ( $\text{MoS}_2$ ) seemed to serve very well as a lubricant for the lunar drill rig. It is a solid lubricant with a temperature range of  $-300^\circ\text{F}$  to  $1200^\circ\text{F}$  in a vacuum. It retains its lubricating properties under increased loads and speed. It can be bonded to both the aluminum and steel parts, as long as curing temperatures are not allowed to damage the materials.

## DRILL RIG ANCHORING

The anchoring device chosen for the drilling rig is an auger anchoring system. Once the auger is in place, the amount of force required to pull the auger out is defined by the type of soil and the angle of repose. The angle of repose is an angle measured from the shaft of the auger. The value of this angle is determined by the composition of the soil. The lunar top soil is a dry-cohesive material with a density of  $187.43 \text{ lb/ft}^3$ . For this type of material, the angle of repose varies between  $30^\circ$  and  $40^\circ$ . For the design of the auger system, a value of  $30^\circ$  was used for the angle of repose, since this value gives the minimum value for the volume of the cone.

The maximum force that the anchoring system needs to restrain is  $2000 \text{ lb}_f$ . Excluding the weight of the drill rig, each anchor leg would need to withstand an upward force of 667 pounds. Using a safety factor of 33.3 %, each auger needs to restrain 1000 pounds upward force.

Using these design parameters, it was determined that an auger 1.5 meters long would be sufficient to withstand nearly 1300 pounds of upward force. This exceeds the design values by almost 300 pounds.

Installation of the anchor system is done in a similar fashion as the core drilling. The end of the anchor shaft is designed similar to the end of the drill string shafts. Instead of the auger shaft assembly, place the anchor auger on the end of the drill string. Using the operation

instructions for core drilling, insert the three augers at the positions where the drilling rig's legs will be placed. After the auger is drilled into the ground, remove the pin from the shaft and move the rig to the next position for drilling. To remove the augers, connect the drill shaft to the auger shaft, and reverse the drilling process.

Calculations and specifications for the auger anchoring system are given in Appendix C.

## DRILL BIT AND SHAFT

### Introduction

It is desired to design a drill bit, chip removal system, core retrieval system, and drill string to suit a lunar drill rig. The bit must cut through the lunar crust to a depth of 100 meters. During the drilling process, core samples must be retrieved and rock particles must be transported from the drilling environment.

Previous lunar drilling has been conducted, however on a small scale. This drilling has involved only hand drills and depths up to one meter. However, NASA has conducted several experiments on Earth which simulate the lunar environment. Previous data from these experiments has been obtained and is used extensively in this project. Several empirical results from NASA studies were used due to the complexity of the analytical approach.

A number of environmental considerations were imposed on the project. The factors include near-zero absolute pressure, low ambient temperature, and low gravity. The near-zero pressure imposes a non-conductive (thermal) atmosphere restriction. Low temperature and gravity are expected to be an advantage to a drilling scheme. Along with environmental considerations, lunar soil characteristics were considered. Minerals expected include basalt, granite, marble and limestone. Basalt was used by NASA for optimization of drill bit design.

Several difficulties inherent in the lunar environment

are expected. Bit cooling due to convection is impossible, therefore an alternate cooling scheme must be devised. Also, chip removal must be achieved without the aid of liquid or foam streams as is done on Earth. Another consideration is particle adhesion. In a low-pressure environment, boundary layers around particles are absent, therefore, direct adhesive forces dominate and result in accumulation of particles and increased frictional forces.

## Diamond Bit Design

After consideration of environmental factors, it was decided that the bit is the most highly constrained component of the down-hole system. A diamond/matrix bit was chosen because of its durability and high temperature characteristics, and because it is self-sharpening. In order to produce interchangeable bits, the matrix is to be bonded to AISI 1040 CR steel. The steel is machined to interface with the auger/core removal component.

The most significant design consideration for the bit is operating temperature. Additional parameters to be considered are hole size, core size, chip channel dimensions, diamond density within the matrix, and cutting face geometry. Specific consideration was taken for optimum bit life and mechanical chip removal (instead of fluid chip removal).

The maximum hole diameter was chosen to be 90 mm and the core diameter was chosen to be 60 mm. These dimensions are a balance between a hole too small to be useful for core removal and a hole too large to drill. Twelve chip channels were used as a compromise between maximum cutting surface and chip removal efficiency. The width of a channel is two millimeters which is expected to be larger than a chip diameter. Using a mean radius of 37.5 mm, the circumference of the bit face is 235.6 mm. Subtracting the accumulated width of 12 channels at two millimeters each from the circumference yields a useful cutting length of

211.6 mm, or 17.6 mm per tooth.

Using a diamond/matrix scheme with 100 diamonds/carat uniformly distributed throughout a sintered metal matrix, approximately eight diamonds/tooth will be exposed at all times. Since experimental use of diamond bits has indicated that the optimum drill rate can be achieved when a thrust of 4-32 lb<sub>f</sub>/stone is applied, using eight diamonds/tooth, the force/diamond will vary from 5-21 lb<sub>f</sub>/diamond as thrust is varied from 500-2000 lb<sub>f</sub>. This will require five carats/bit. The diamonds to be used are West African Bortz class AAAA diamonds because of their superior cutting ability. The matrix to be used is a type 4 powdered metal available from the Christensen Diamond Products Company.

The face of the bit is rounded to avoid extreme temperatures that would be encountered at the edges if a square face cutter was used. Relief is provided above the plane of maximum tooth thickness to reduce frictional area. Chip channels are used to provide escape for chips from the cutting surface to the chip removal system.

Before the final design was chosen, two alternate designs were considered. First, a standard square-toothed bit with one layer of pre-arranged diamonds was examined. This scheme would yield excellent chip removal but poor durability. The second alternate design was a standard carbide-tipped cutter which also proved to be less durable than the diamond/matrix bit. The final design yielded the best compromise between long life and chip removal.

## Chip Removal System

Once rock chips are produced, they must be mechanically removed from the drill face. Since fluids may not be used in the lunar environment, an auger system was chosen. Augers are only required on the leading sections of the drill string. Clearance between the hole and the auger leads must be provided to avoid excessive frictional loading on the shaft. The pitch angle should be chosen to remove chips quickly to avoid chip accumulation near the bit. The total volume of the auger flights must equal or exceed the volume cut during each incremental drill run. The augers must also be sufficiently smooth, to avoid adhesion of chips, and abrasion resistant to maintain smoothness.

The final design of the auger shaft was based on experimental results of previous NASA studies. The use of three auger leads pitched to 15 degrees was found to be the most efficient arrangement. To sufficiently accommodate chip flow, a three millimeter auger lead depth and width was used. Since two meters was chosen as the maximum continuous run, the augers must be used for the first eight meters of the drill string. Due to its high abrasion resistance, titanium-nitrate plating was chosen to cover the augered shafts.

The transport of chips from the bit face to the auger flights is accomplished using twelve chip channels leading to the three flights. These numbers were chosen from

experimental tests performed on simulated lunar soil. The chip channels are built into the bit and are therefore made of matrix material.

A system to remove the core after a drill run is necessary for the requirements of the project. The removal system could be incorporated in the drill string or could be a separate mechanism. It must grip and break the core in order to remove it from the hole.

The system chosen is an inline component placed directly above the bit in order to maximize core length. It is an upward-movement actuated collet-type mechanism consisting of four knurled jaws mounted in an tapered recess. They are restrained to only vertical motion by pin-slider mechanisms. As the drill string is retracted from the hole, the knurled jaws slide down the angled recess and grip the core, and the grip strength increases with increasing upward pull. The maximum upward pull will occur at the 100 meter depth, and is due to tensile strength of the core and drill string weight. This value is not expected to exceed 8000 lb<sub>f</sub>, and is accounted for in the design of the drill string.

A core removal system that is not incorporated in the drill string was considered but was decided to be inefficient because the drill string apparatus would have to be removed in order to use this system. This alternate system was a collet arrangement suspended from a cable which is lowered into the hole once the drilling apparatus has been removed. The collet will clamp at the base of the

core sample, and then, the sample is retrieved by withdrawing the cable.

A design incorporating both the auger system and the core removal system is needed to conserve space and weight. The major problem foreseen in this integrated design is the manufacturing of the auger/core removal system from one piece of stock. Since the auger shaft must be hollow, thus suggesting tubular stock, and the core removal system is of different inside diameter, two pieces appear to greatly simplify manufacturing.

The final design chosen involves welding the auger shaft to the core removal section. The outside of the core removal section must be timed to interface with the auger shaft. These components, when welded, act as one system.

Since the overhead clearance of the drill rig is four meters, two four-meter auger shafts are used. The first section houses the core removal system, and is connected directly to the drill bit. The second section is timed to interface with the auger leads of the first section. The upper end of the second auger section interfaces with the first standard section of the drill string.

## Standard Drill String Section

A drill string made of sections no longer than the overhead clearance of the drill rig is required. The primary consideration is light weight, but several physical limitations are imposed by the drilling process and rig capabilities. The variable parameters involved are outside radius, wall thickness, length, flex rate, and material properties of the shaft. Constraints imposed by the drilling process are maximum applied torque and maximum tension and compression. A minimum safety factor of two was chosen for torsion, tension/compression, shear, and buckling.

In order to minimize weight and satisfy all safety factor and size constraints, a spreadsheet parametric analysis was used. A maximum torque of 126 ft-lb<sub>f</sub>, a maximum compression of 2000 lb<sub>f</sub> and a maximum tension of 8000 lb<sub>f</sub> are required in a worst case drilling situation. The outside radius, wall thickness, length and flex rate were varied to produce the combination of least overall weight. ASTM 2024-T4 Aluminum was chosen as the drill string material because of its high strength-to-weight ratio. Drill string weight varied from over 1200 lb<sub>f</sub> to less than 600 lb<sub>f</sub> for acceptable combinations of variable parameters.

The minimum weight solution has a radius of 39 mm, a wall thickness of 4 mm, a length of four meters and a flex rate of 0.6 degrees/meter. The string weight for this

combination is 567 lb<sub>f</sub>. The auger section weight is 37 lb<sub>f</sub> for a radius of 44.5 mm, a wall thickness of 3 mm and a flex rate of 0.6 degrees/meter. The total lunar weight is 100 lb<sub>f</sub>, thus the total shipping weight is 604 lb<sub>f</sub>.

## Component Interfaces

A mode of interfacing the bit, augers, drill string, and motor is desired. Each interface must transmit torque, and tensile and compressive forces. The interfaces should be simple in design and operation, yet meet the design parameters of the other components of the system.

The specific design involves five different component interfaces. They are the bit/auger, auger/auger, auger/string, string/string, and string/power head. The bit/auger interface must allow for interchangeability of bits and cannot increase the overall diameter of the bit. The most logical solution appears to be right-hand machine threads. Therefore, M60-6 threads are used in the final solution. Ease of interchangeability was sacrificed in order to meet the diameter requirement. Pins or set screws are not necessary because the rotation of the shaft is opposing the back-out direction of the threads.

An auger/auger interface is needed because the overhead clearance of the rig is half the length of the augered section. The major constraints for this interface are maximum diameter and ease of breakdown. Since the clearance between the hole and body of the auger shaft is 3.5 mm, the interface cannot be external. A male/female internal joint with three threaded pins (set screws) was chosen as the final solution. A threaded interface, like the bit/auger connection, is not feasible because the connection must be broken after each drill run of two

meters.

The transition from the auger to the standard string section is accomplished by the auger/string interface. This connection is similar to a standard square-jaw clutch encountered in many power transmission systems. Four "jaws" transmit torque and compressive force. A sleeve encases the "jaws" to prevent buckling. The sleeve is welded to the "upper" end of the auger and a pin is inserted through the sleeve and the "lower" end of the standard section. This pin transmits tensile forces through the coupling. This interface is different from the string/string connection only because the auger is larger in diameter than the standard section. The string/string coupling uses the same "four-jaw" mechanism with a sleeve and pin as the auger/auger joint. The same scheme is used to couple the drill string to the power-head.

## Thermal Considerations for Geometric and Physical Constraints

A major consideration in the drilling process is dissipation of thermal energy. Standard drilling practice dictates the use of liquid or gaseous coolants to prolong bit life. However, the lunar environment prohibits the use of materials that evaporate in the near-zero ambient pressure. Therefore, dry drilling is required and bit life is a strong function of the mechanical mode of heat transfer.

Approximately 2/3 of the heat generated down-hole is produced at the bit. According to NASA's previous experimentation with drilling in a simulated lunar environment, 80% of the heat generated at the bit remains in the chips produced. Since the thermal conductivity of the chips is approximately 0.0007 B/hr-ft-°F, they retain this thermal energy and it does not present itself as a problem in bit or shaft temperature. However, the 20% that is transferred into the diamond/matrix must be dealt with.

Under the most extreme conditions (126 ft-lb and 2000 lb<sub>f</sub> down-force at the bit), 12,600 B/hr is generated, therefore, 2,520 B/hr must be transmitted up the drill string. Experimental results show that the bit temperature at this extreme will be no more than 500 °F. Using a Class 4 powdered metal matrix and AISI 1040-CR steel, the temperature, by pure conduction, at the bit/auger interface is 455 °F. A constraint imposed on the drill string is a

maximum temperature of 70 °F. However, the auger shaft is designed to accept 455 °F and deliver no more than 70 °F to the standard drill string sections. Incidentally, the steady-state temperature will be 70 °F, 1/6 of a meter from the bit.

Since temperature is a major factor in bit life, temperature feedback was thoroughly investigated. The maximum safe operating temperature for a diamond/matrix bit is 1200 °F. NASA's previous investigations have proven that a bit will be considered "dead" when a constant feed rate can no longer be maintained at a down-force of less than 2000 lbf and a rotational rate of 150-400 RPM. At this point the maximum bit temperature will be under 500 °F. This is well within the operating range of the materials used in the design. Temperature feedback for the bit was therefore deemed unnecessary due to close correlation between maximum bit temperature and down-force. A qualitative relationship between bit temperature and down-force has been theorized and may be quantified by experiment only. A graphical relationship of down-force vs. % wear along with heat transfer calculations is presented in Appendix D.

## Mechanical Considerations for Proposed Geometric, Physical and Thermal Constraints

Quantitative mechanical considerations of the down-hole system are governed by the qualitative restraints and decisions that were realized early in the design process. Unlike thermal characteristics, the mechanical analysis is independent of the research formerly conducted by NASA. The analytical approach is straight forward and is sensitive to four controllable parameters. Three are geometric properties (shaft radius, wall thickness and length) and the other is intrinsic material properties.

Mohr's Circle was used to design against tensile, compressive, and shear failure. Euler's Equation for buckling was chosen over the J. B. Johnson Formula to design against buckling due to the ratio of length to radius of gyration. These equations were incorporated into a "Symphony" spreadsheet, in order to document the effects of varying the four pertinent design parameters. Several different sets of the four parameters along with their resulting effects are presented in Appendix B, with the final design parameter set highlighted.

Twist in the drill string could become a problem if the flex rate is not sufficiently controlled. Torsional vibrations could also arise due to a drill string that is not stiff enough to damp out instantaneous deflections. A maximum flex rate of 0.6 degrees/meter was chosen for the final design. This is based on verbal recommendations by

Mr. David Federer and a reference book on vibrational considerations.

The maximum power required to drive the down-hole assembly is 7.6 HP. Of this total, 4.9 HP (determined analytically) is required to drive the bit, and the remaining 2.7 HP (determined experimentally by NASA) is used to drive the auger/chip removal system. Detailed calculations of power requirements are presented in Appendix D.

## Conclusions

Mechanical and thermal properties of available materials indicate that a feasible solution is obtainable. Design specifications leading to safety factors of two or more were obtained for the entire down-hole system. Solutions to thermal design problems have been based on previous NASA experimentation conducted in a simulated lunar environment.

## Recommendations

It is recommended that a prototype of the sub-surface apparatus be constructed and tested in a simulated lunar environment in order to validate critical data from NASA studies. A testing phase would also allow for further optimization and validation of the operating characteristics of the bit and auger system. During this test, the qualitative approach to bit-wear determination can be quantified. Following the testing phase, honing of the design parameters can be implemented for use in the final design.

## RADIANT COOLING SYSTEM

Operation of the drilling device in the lunar environment presents unique problems in heat dissipation. Three electric motors, which will be required to run continually for  $n$  hours to complete the task, will produce heat on the order of  $0.192 \text{ W}/_{\text{m-}^{\circ}\text{C}}$ . If this heat is not channeled away from the thermally sensitive motor parts, premature failure will occur. Because of the expense and inconvenience of repairs on the moon, extension of equipment life is important. Therefore, a system to insure operation at temperatures around  $276^{\circ}\text{K}$  should be implemented.

The electric motors will be enclosed in a vessel filled with hydrogen pressurized at 15 psi. A compressor is required to provide the pressurized atmosphere. Oxygen, nitrogen and hydrogen were investigated as potential working fluids because of their capacities of heat transfer. Hydrogen was picked over the others for its adequate performance characteristics along with the fact that it generates the least drag on engine performance.

Heat is produced in the motor at the stator and armature and must be removed. This is accomplished by passing the fluid through the motor housing. The heat is carried away from the point of generation by conduction and convection, and amounts to 2478 W.

As the fluid passes over the motor parts, friction is produced, causing the motor to work harder and produce more heat. Hydrogen will minimize this effect on the motor performance because of its low drag factor.

The heated fluid exits the motor housing and enters a shielded pipe. This pipe transports the fluid to a system of cooling tubes. 150 aluminum tubes, with a diameter of 0.15 meters and a length of 0.30 meters, make up the cooling array. The array is enclosed in a rectangular box of dimensions 1 m by 8 m by 0.30 m.

The fluid transfers heat to the tubes, which are cooled down. The top of the enclosure is attached directly to the cooling tubes and is a flat surface composed of pure aluminum. The heat absorbed by the tubes is conducted directly to the flat surface where radiation occurs.

Radiation is closely related to the absolute temperature of the radiating surface. The quality of actual radiation depends on this temperature and the properties of the material coating the surface.

The quantity of radiant heat transfer in space is directly related to heat received and heat rejected. By directing this surface toward deep space no sources will face the radiator and no absorption occurs. Although some energy will be received from deep space, it is of sufficiently small magnitude to be considered negligible. A solar screen may be needed during the lunar day to shade the radiating surface from heat sources, such as the sun.

Radiation occurs as a function of the Stephan-Boltzman constant ( $\sigma$ ), the area, and the fourth power of the temperature.

$$Q = \sigma AT^4$$

In order to dissipate the heat generated by the motors an area of 8 square meters is required.

After contact and transfer with the cooling array, the fluid exits the radiator section at a lower temperature and again enters the shielded pipe. This transports the fluid back to the motor housing, where the process repeats itself.

Insulation is placed around all surfaces, except the radiating surface, to ensure that unwanted heat is not absorbed from surrounding sources.

Specifications and calculations for the radiant cooling system are located in Appendix F.

## Recommendations And Conclusions

Several systems were investigated for possible use for heat dissipation in the lunar environment. These are: a system of subterranean heat exchanger pipes, a system of disposable heat sinks, and a device known as a Liquid Droplet Cooler. For various reasons, each was preempted for the system described in the previous section.

It should be noted that the droplet cooler shows promise and with further research could fulfill the system requirements. The cooler utilizes the concept of radiation and directly exposes a sheet of millions of submillimeter droplets to space. The droplets are sent on a trajectory from a spray bar, and collected again after sufficient heat transfer has occurred.

This innovative system promises to significantly reduce the mass required for current radiating systems. However, in order to become operational, uncertainties existing in droplet collection and surface rewetting must be addressed. More research is suggested, with emphasis on ionic charging as a mode of recollection.

The subterranean system also shows promise, however, not enough is known of the subterranean lunar environment. Thermal conductivity of lunar dust is on the order of  $0.0015 \text{ W/m-}^\circ\text{C}$ . This is much too low for efficient utilization. If more favorable conditions are encountered several meters below the surface, a subterranean system similar to those used to subsidize heat pumps in residential applications on

Earth, could be justified.

Due to simplicity of operation and projected reliability, the system utilizing the hydrogen medium and the aluminum radiator is to be implemented. Maintenance is not recommended at the sight, so provisions exist for detachment of particular modules, and transport to a lunar base for repairs, if necessary.

Periodic examination of the system is recommended to ensure that corrosion has not occurred in amounts sufficiently large enough to impede operation.

## FINAL CONCLUSIONS

Drilling to depths of 100 meters is feasible in the lunar environment, utilizing the previously described tripodal drill rig. The system is lightweight and semi-modular, affording relative efficiency in transport to the moon. Once in the field, portability and durability are inherent in the design to ensure productivity and long life.

The tripodal structure provides support of the system and controls the angle of entry into the lunar terrain. Although the structure is inherently stable, an anchoring system at each leg of the structure increases stability, and resists torsional forces transferred through the shaft.

The diamond matrix bit produces a bore of 90 mm, while generating a core sample 60 mm in diameter. This core will provide basic information concerning the formation of the lunar substructure, and will determine whether mining operations should continue at the sight.

To drive the entire system, two 240 volt, 10 hp DC motors are required. Heat generated by these motors will be dissipated by radiation.

## RECOMMENDATIONS

This design represents the careful considerations directed at solving the unique problems of lunar drilling. All major areas of design were addressed. However, this study should be continued in hopes of further optimization of the system.

It is recommended that either a prototype of this model or a modified version be constructed. Testing the design in simulated work conditions will serve to reinforce and substantiate conclusions while exposing any overlooked constraints.

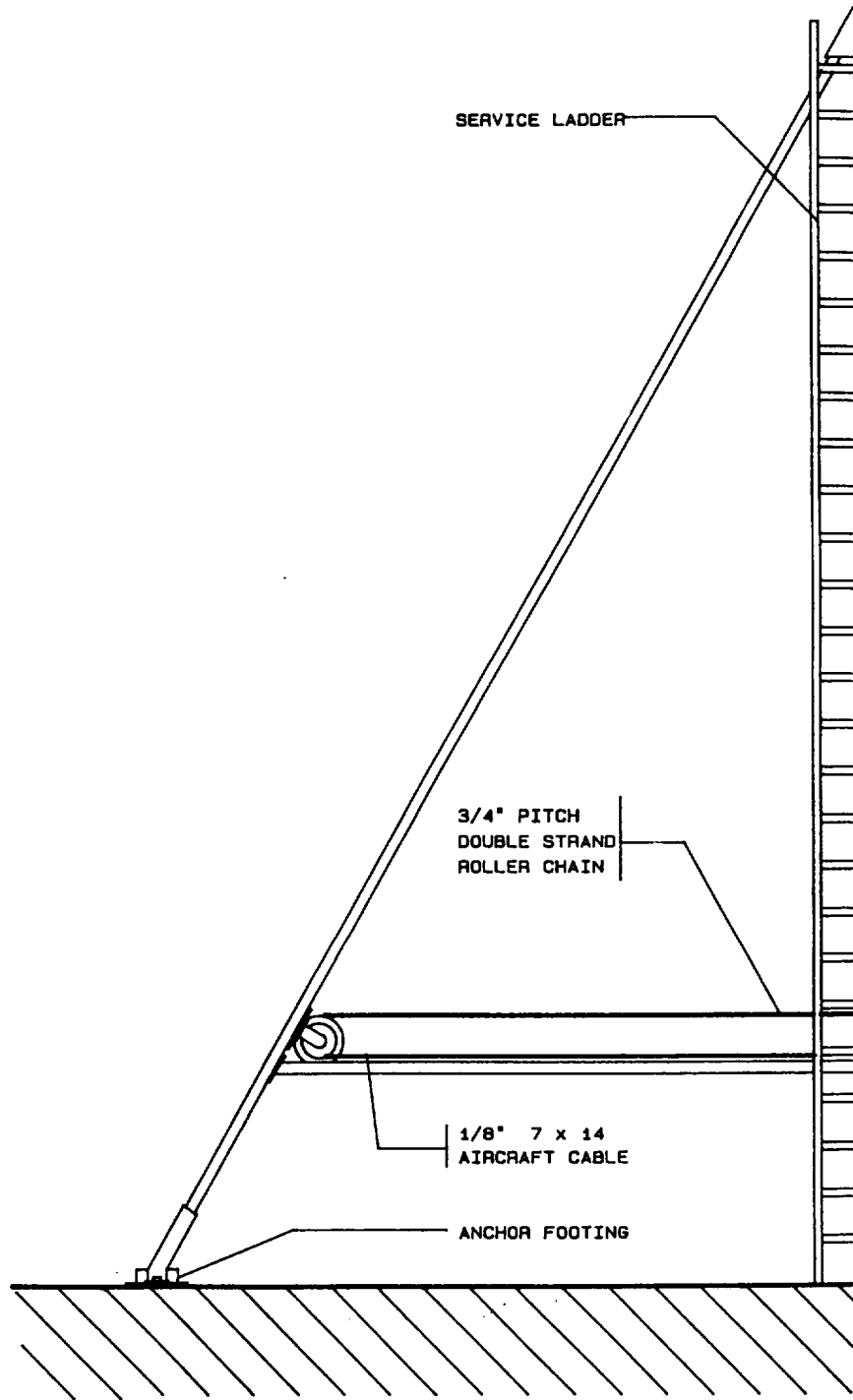
## INSTRUCTION MANUAL

The following is a sequential list of operating instructions for the proposed design. This sequence follows installation of the anchor system.

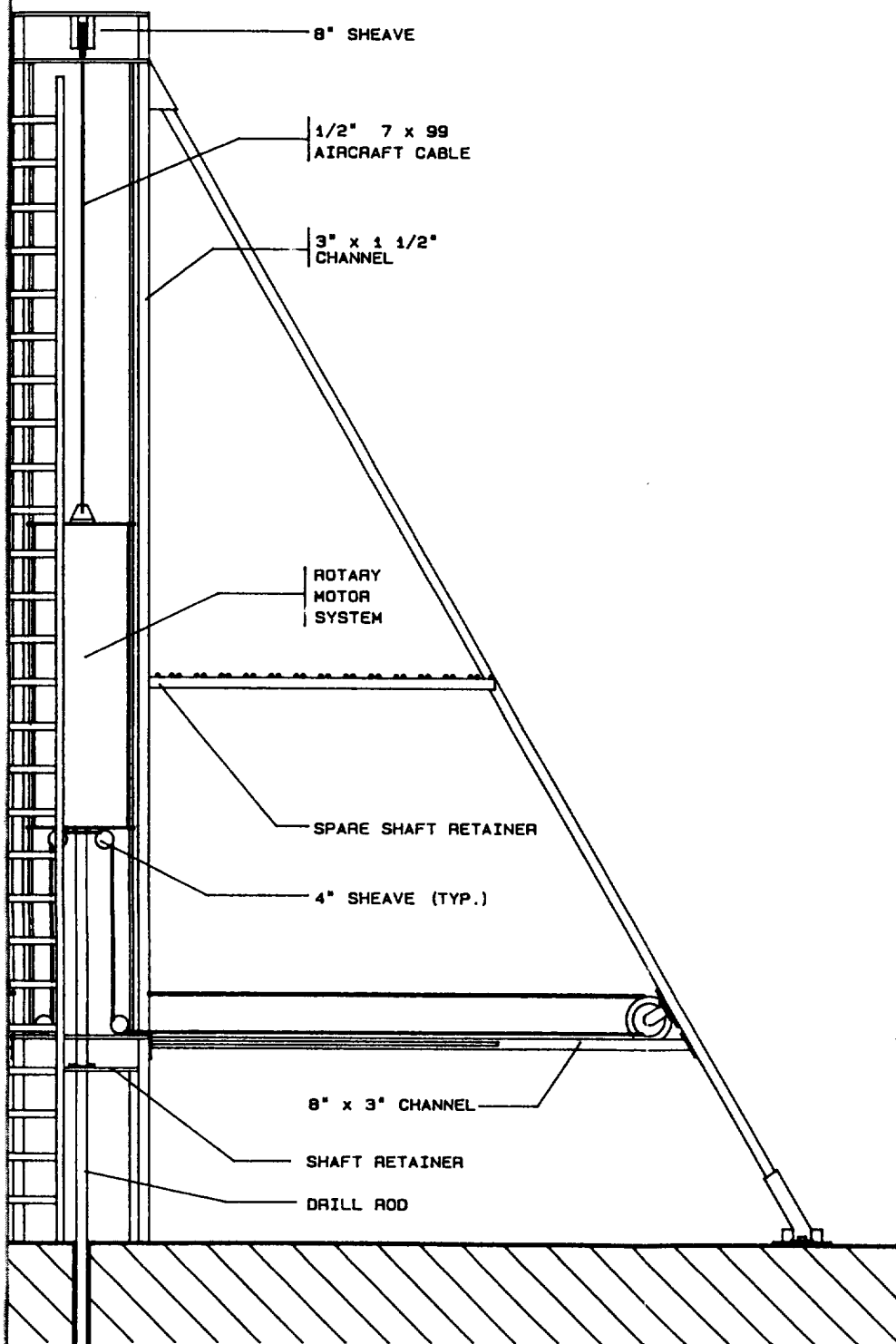
1. Connect the bit/auger shaft to the powerhead.
2. Begin drilling at 210 RPM clockwise and a constant feed rate of 2.0 inches/minute.
3. Using position feedback, drill 2 meters deep.
4. Stop rotational and linear motion of the powerhead.
5. Retract the string thereby breaking the core.
6. Resume rotation and withdraw the drill string.
7. Disconnect the auger shaft from the powerhead.
8. Remove the 2-meter core sample from the "top" end of the bit/auger shaft.
9. Repeat steps 1-8 (one time).
10. Replace the bit/auger shaft into the shaft retainer at the top of the hole with the auger/auger interface exposed.
11. Connect the second auger shaft to the first with set screws.
12. Release the shaft retainer and lower the drill string into the hole.
13. For each section to be added to the drill string, clamp the down-hole drill string with the shaft retainer and connect the next shaft using the sleeve/pin connector.

\* NOTE: Special caution must be exercised when

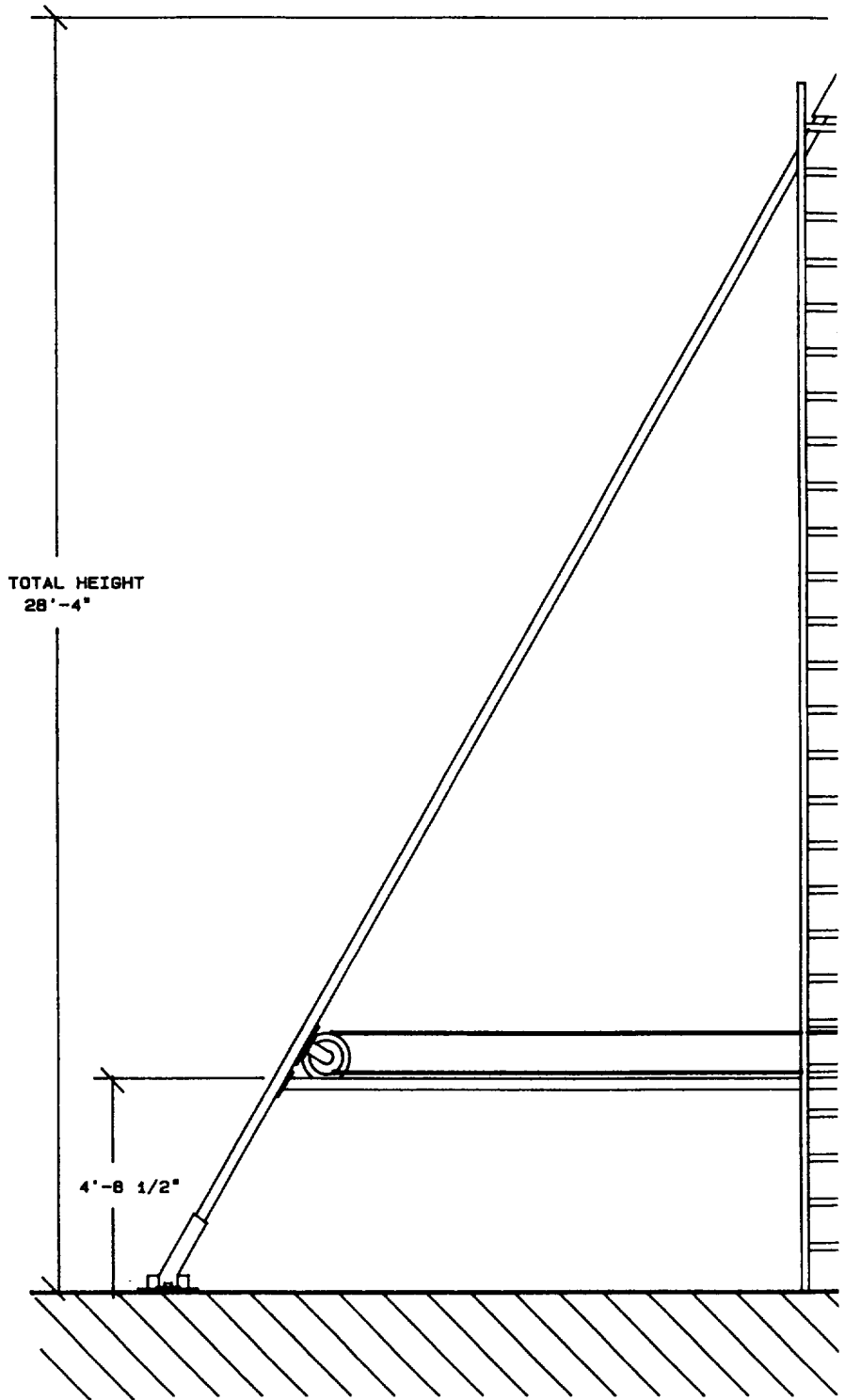
- replacing the drill string to prevent "dropping" the string into the hole with too short a shaft.
14. Connect the upper-most section of the drill string to the powerhead.
  15. Drill at 210 RPM clockwise and a constant feed rate of 2.0 inches/minute for 2 meters.
  16. At the end of a 2 meters stroke, stop the rotation and linear motion of the powerhead and retract the string slightly to fracture the core.
  17. Restart the clockwise rotation.
  18. Raise the drill string until the top 4-meter section is exposed.
  19. Stop the upward motion and rotation of the powerhead.
  20. Clamp the lower drill string in the shaft retainer.
  21. Disconnect the top section from the powerhead and lower drill string.
  22. Store the removed section in the shaft section rack.
  23. Lower the powerhead and connect it to the next section of the drill string.
  24. Repeat steps 17-23 until the entire drill string has been removed.
  25. Remove the core from the "top" end of the bit/auger shaft.
  26. Repeat steps 10-25 until the desired depth has been reached.



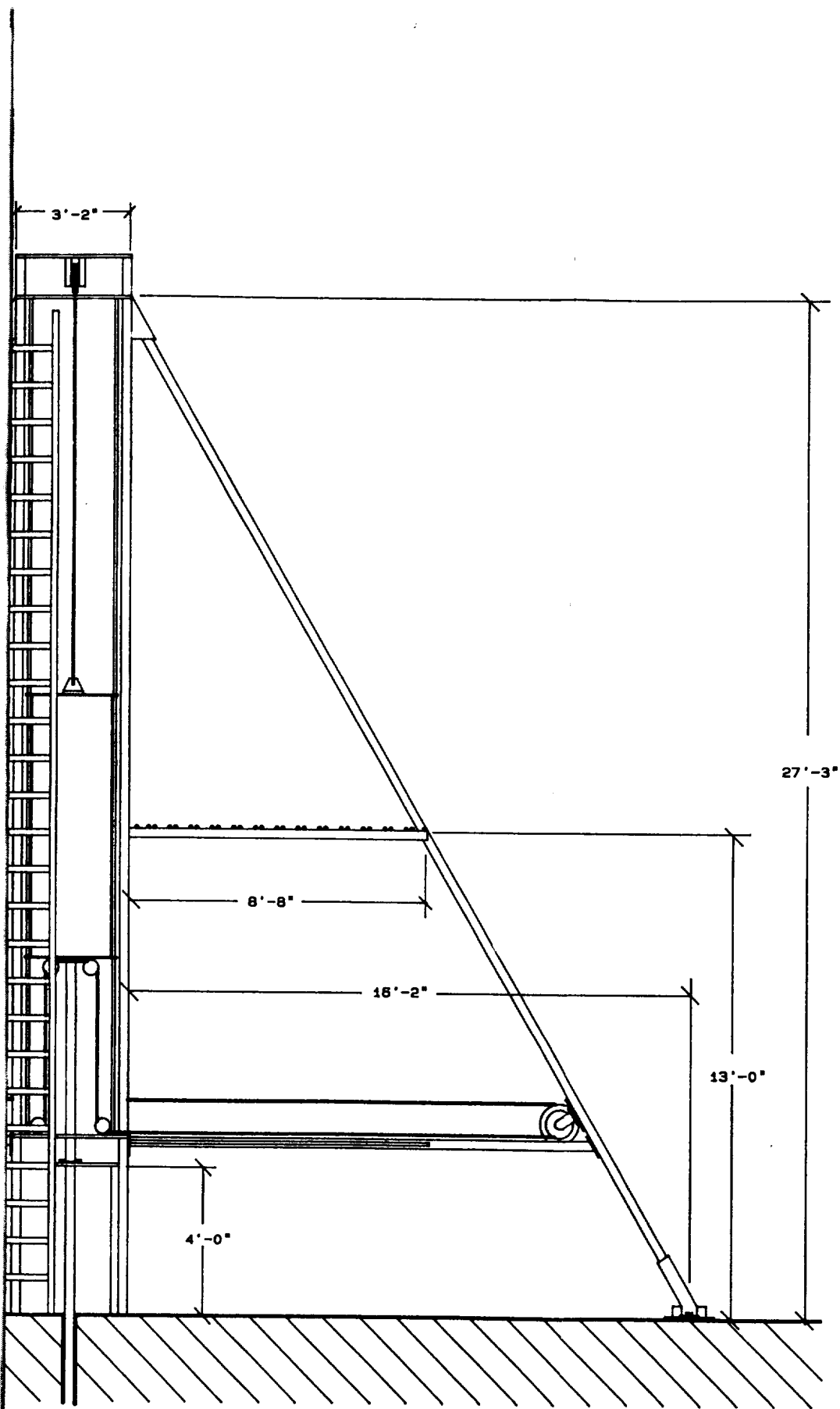
FOLDOUT FRAME



FOLDOUT FRAME 2

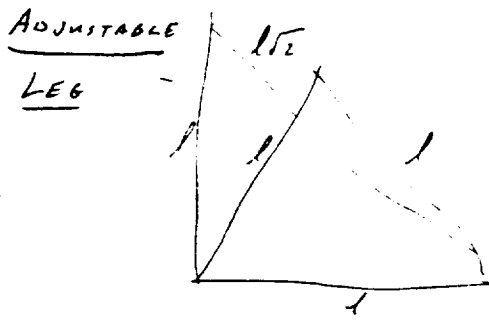


FOLDOUT FRAME |



FOLDOUT FRAME 2

# APPENDIX A



$$\Delta L = l(\sqrt{2} - 1)$$

$$l = 340 - 13 = 327$$

$$\Delta L = 135.45$$

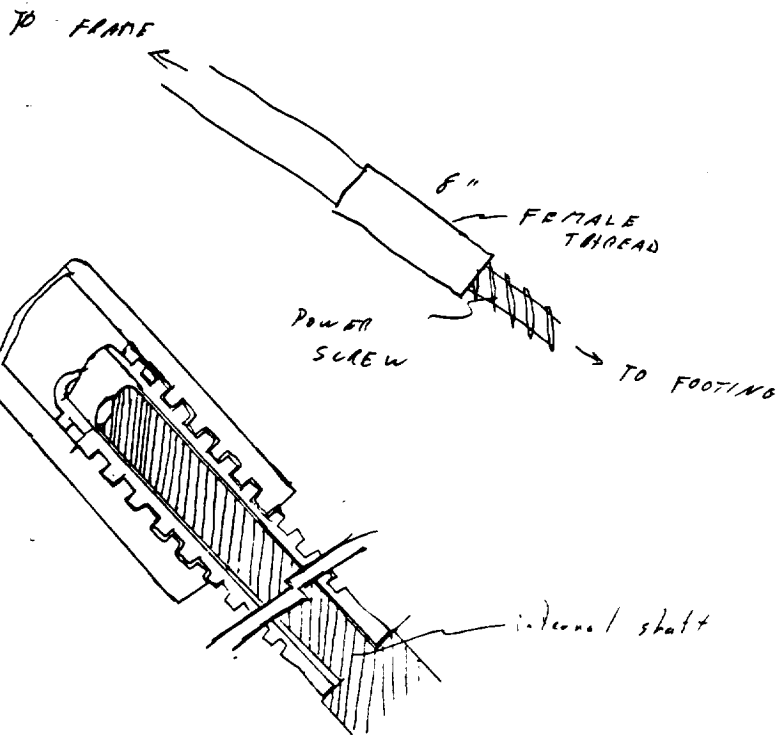
$$\text{use } \Delta L = 145 \text{ in}$$

using

POWER SCREW:

NEED 145 in of threaded section

LOCATE THREADED SECTION AT BOTTOM OF ~~THAT~~ ADJUSTABLE LEG



using EULER'S EQN

$$P_{cr} = \frac{C \pi^2 E I}{L^2}$$

$$P_{cr} = 2000 \text{ lbs}$$

$$\Rightarrow I = 0.3447$$

$$\therefore \text{tube dia.} = 2 \times \frac{1}{4}$$

$$(\text{@ } I = 0.537)$$

$$\text{internal shaft dia.} = 1 \frac{1}{2} \text{ \"}$$

with  $\frac{1}{4}$  high threads

$$\text{I.D. Leg} = 2 \frac{1}{2} \text{ \"}$$

# APPENDIX A (cont'd):

WITH

$$ID. Lr = 2.5 \text{ inch}$$

&

$$I = 0.3447$$

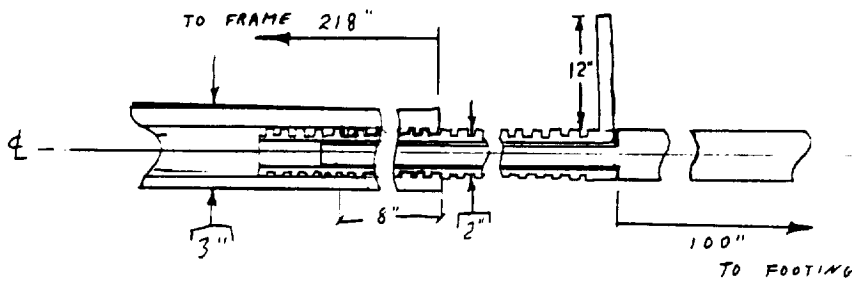
$$I = \frac{\pi}{64} (d^4 - d_i^4) = 0.3447$$

$$\frac{\pi}{64} (d^4 - 2.5^4) = 0.3447$$

$$d^4 = 46.0648$$

$$d = 2.6055$$

use  $d = 3''$



218" : 3" x 1/4" ALUMINUM TUBE

100" : 2 1/2" DIA. ALUM. SHAFT

W/ 145" LONG 1 1/2" SHAFT FOR SUPPORT

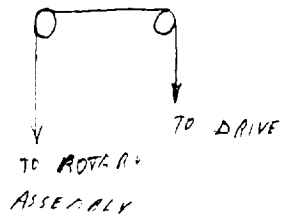
WELDED TO UPPER END

145" : 2" x 1/4" TUBE W/ EXT THREADS TO SERVE AS  
POWER SCREW.

## Appendix A (cont'd)

### DRIVE RATIOS

Since 1 motor controls both raising & lowering of the motor the cable feed must be equal & opposite.



DURING DESENSATION A REFEED OF 2.0 in/min of cable must be maintained.

$\omega_D$  = RPM OF VERTICAL DRIVE MOTOR

$\omega_S$  = RPM OF INDIRECT DRIVE SHAFT

$N_P$  = # of teeth on pinion of double strand roller chain

$N_S$  = # of teeth in sprocket of double strand roller chain

$$n = N_S / N_P$$

$$\omega_D = \frac{N_S}{N_P} \omega_S$$

$$= n \omega_S$$

ITERATIVE:

To maintain a 2.0 in/min feed using a 6" DIA SPOOL for the CABLE

$$\omega_S = \frac{2.0 \text{ in/min}}{6 \text{ in}}$$

$$= 0.1061 \text{ rev/min}$$

$$\Rightarrow \omega_D = 0.1061 n \text{ rev/min}$$

With 50:1 gear reduction on motor and motor running at 50 rpm

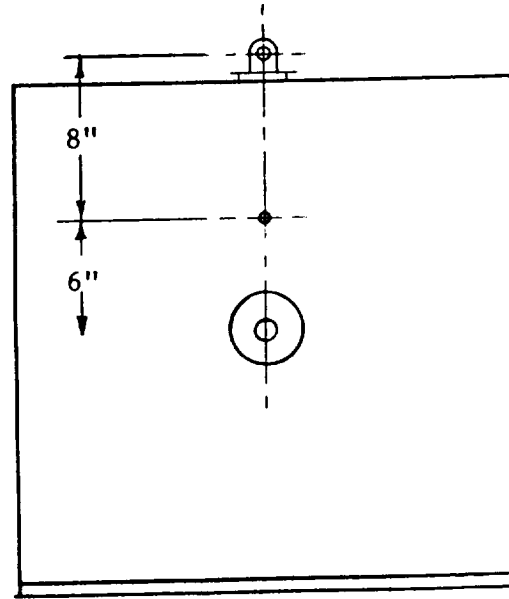
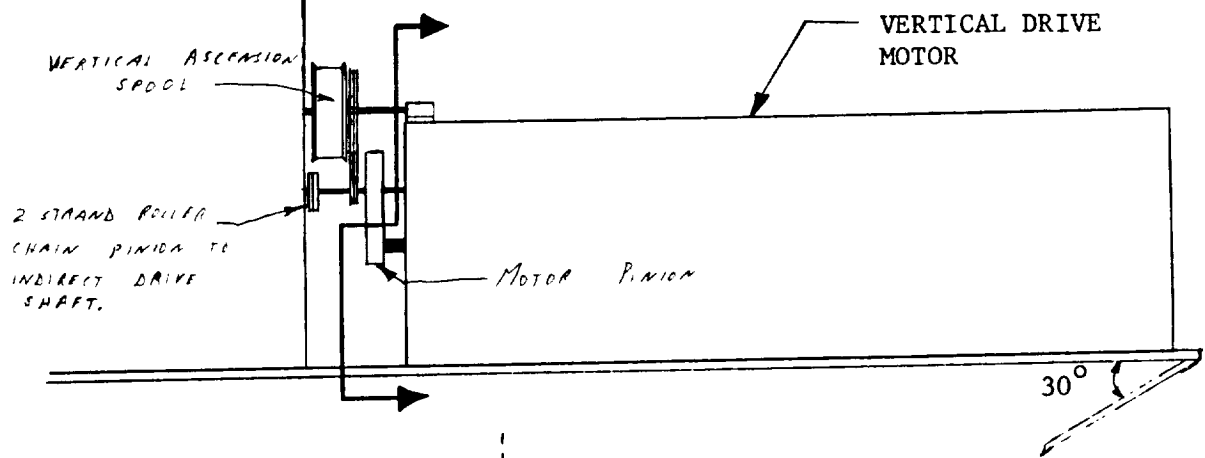
$$\Rightarrow 0.1061 n = \frac{50}{50}$$

$$n = 9.4248$$

$$\text{LET } n = 10$$

$$\Rightarrow \omega_D = 53.0516 \text{ rpm} \quad \text{to achieve 2.0 in/min of cable feed}$$

APPENDIX A (cont'd)



## APPENDIX A (cont'd)

Using  $\omega_D = 53.0516 \text{ rpm}$

Determine  $d_v$  = spool diameter for ascension cable

Since Feed is  $2.0 \text{ m/min}$

$$2.0 \text{ m/min} = d_v \pi \text{ rpm} = 53.0516 \text{ rpm} \cdot \pi \cdot d_v$$

$$\Rightarrow d_v \pi = 0.0120 \quad ; \quad \pi \text{ is constant diameter to } \pi$$

$$\text{Let } \pi = \frac{1}{10} \quad \text{if } 10:1 \text{ gear ratio}$$

$$\Rightarrow d_v = 6 \text{ inch}$$

if  $\pi = \pi$  with  $50:1$  gear reduction still present

All spools are 6 inch in diameter.

### TIME CONSIDERATION

For WITANE OF DRILL STRING, use TOP RATED SPEED:  $900 \text{ RPM}$

$$\frac{\text{Ascension}}{\text{Desension}} = \frac{18}{1}$$

$$\rightarrow \text{Time, descent for } d_m = \frac{d_m}{2 \text{ m/min} \cdot \frac{7.5 \text{ m}}{10}} = 78.7407 \text{ min}$$

$$\rightarrow \text{Time, ascension for } d_m = \frac{78.7407}{18} = 4.3745 \text{ min}$$

Time to bore 100 m hole and remove final core:

$$1 + 2 + 3 + \dots + 24 + 25 = 12.25 = 300 \text{ down strokes}$$

+ 300 up strokes

$$\begin{aligned} \Rightarrow 300(78.7407 \text{ min}) + 300(4.3745) &= 24930.41 \text{ min} \\ &= 415.51 \text{ hrs} \\ &= 17.315 \text{ d} \end{aligned}$$

trouble this due to the human factor

To drill 100 m hole: 52 days maximum.

This time will drop as the crew becomes

more and more experienced

# APPENDIX A (cont'd)

$$P_{cr} = \frac{C \pi^2 E I}{L^3}$$

$$= \frac{1.2 \pi^2 E (64)}{12 L^3}$$

ROTARY MOTOR SLIDE  
ASSEMBLY

$$= \frac{1.2 \pi^2 (10.3)(10^6) 24 h^3}{12 \cdot 66^3}$$

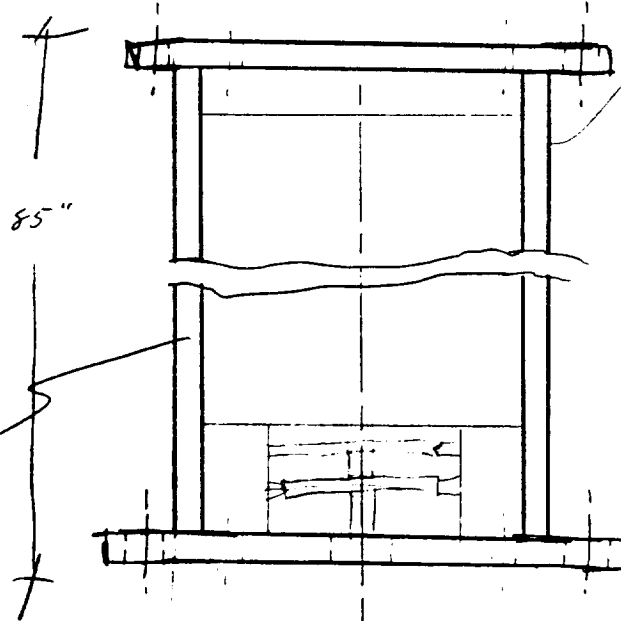
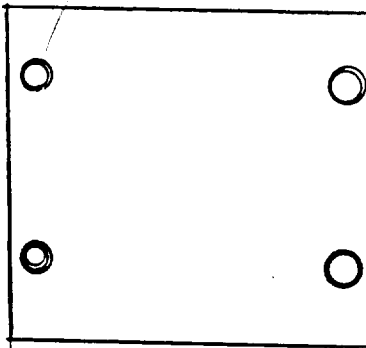
$$h = 0.245274$$

w/5F#4

$$h = 0.38935$$

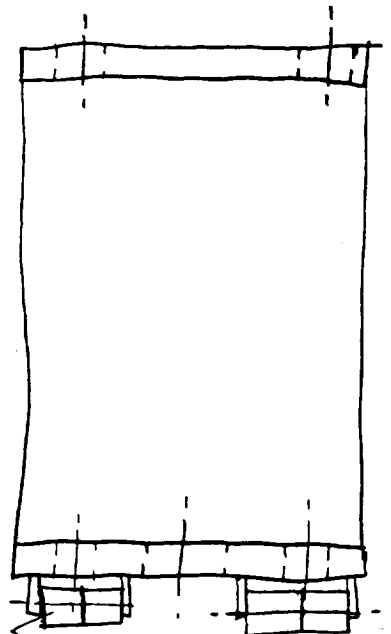
$$\text{use } h = 0.4"$$

1" THRUST BEARINGS  
(TYP. 4 PLACES)



60 x 0.4 x 24

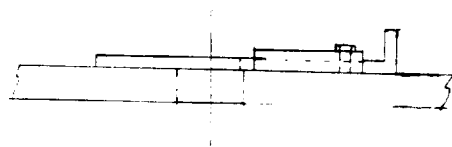
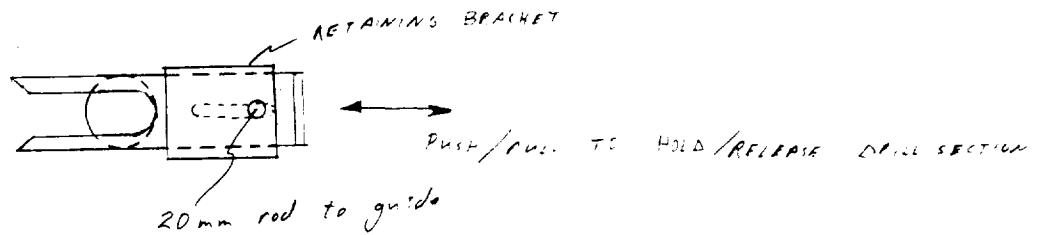
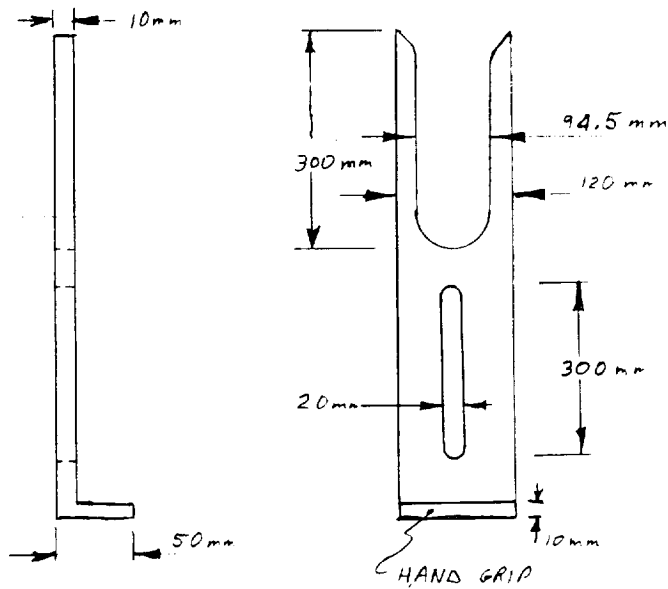
wt: 56.45 lbs



4" sheaves

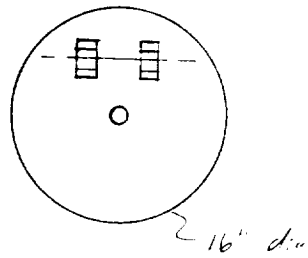
# APPENDIX A (cont'd.)

## SHAFT RETAINER



94.5 mm chosen as inner measure since diameter of shaft is  
 94 mm at point of capture and diameter of rod is  
 93 mm at point of capture.

Appendix A (cont'd)  
Anchor Footing



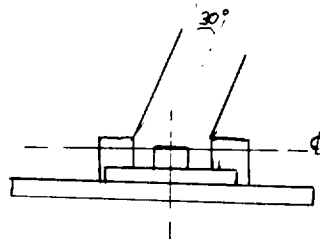
FOOTING



8" DIA  
HOLDING PLATE



SQUARE NUT  
(TO SECURE)



# Dripproof, Fully Guarded **KINAMATIC**® Motors

For Use on M-G Sets or Rectifier Power Supplies †

Type CD

Antifriction Bearings

For Operation in 40 °C Ambient

240, 500, and 550 volts  
Stabilized-shunt or Shunt-wound ★

USE 365 AT

## PRICES†

240, 500-550 Volts (Rectifier Type C or D) ★

Hpt	Speed, Rpm		240 Volts		500-550 Volts	
At Base Speed	Base Speed, Rpm	Rated Top Speed, Rpm	Frame Type CD ‡	List Price GO-2A (240 V) ±	Frame Type CD ‡	List Price GO-2A (500 or 550 V) ±
7 1/2	3500	3500	186AT	\$2168	189AT	\$2516
	2500	3000	L126AT	2516	218AT	2997
	1750	2300	189AT	2997		
	1150	2000	2110AT	3659	2110AT	3659
	850	1700	259AT	4341	259AT	4341
	650	1600	258AT	5250	258AT	5250
10	3500	3500	218AT	2826	218AT	2894
	2500	3000	218AT	2894	219AT	3437
	1750	2300	219AT	3437		
	1150	2000	258AT	4179	258AT	4179
	850	1700	287AT	4995	287AT	4995
	650	1600	327AT	6143	327AT	6143
15	3500	3500	218AT	3447	219AT	3510
	2500	3000	219AT	3510	258AT	4166
	1750	2300	258AT	4166		
	1150	2000	288AT	5124	288AT	5124
	850	1700	327AT	6110	327AT	6110
	650	1600	328AT	7595	365AT	7595
20	3500	3500	219AT	4089	2110AT	4044
	2500	3000	2110AT	4044	259AT	4800
	1750	2300	259AT	4800		
	1150	2000	327AT	5954	327AT	5954
	850	1700	328AT	7304	328AT	7304
	650	1600	366AT	9045	365AT	9045
25	3500	3500	258AT	4557	259AT	4487
	2500	3000	258AT	4487	287AT	5421
	1750	2300	287AT	5421		
	1150	2000	328AT	6852	328AT	6852
	850	1700	365AT	8337	366AT	8337
	650	1600	366AT	10362	366AT	10362
30	3500	3500	287AT	5210	259AT	4932
	2500	3000	287AT	4932	288AT	5981
	1750	2300	288AT	5981		
	1150	2000	365AT	7581	365AT	7581
	850	1700	366AT	9269	366AT	9269
	650	1600	368AT	11408	368AT	11408
40	3500	3500	368AT	13676	407AT	13676
	2500	3000	409AT	16461	409AT	16461
	1750	2100	409AT	20120	508AT	20120
	1150	2000	365AT	9032	365AT	9032
	850	1700	366AT	11037	366AT	11037
	650	1600	368AT	13500	368AT	13500

Hpt	Speed, Rpm		240 Volts		500-550 Volts	
At Base Speed	Base Speed, Rpm	Rated Top Speed, Rpm	Frame Type CD ‡	List Price GO-2A (240 V) ±	Frame Type CD ‡	List Price GO-2A (500 or 550 V) ±
50	2500	2700	327AT	\$6858	288AT	\$6858
	1750	2100	365AT	8085	329AT	8085
	1150	2600	366AT	10422	366AT	10422
	850	1700	368AT	12812	368AT	12812
	650	1600	409AT	15525	407AT	15525
	500	1500	409AT	18671	504AT	18671
60	2500	2700	365AT	8004	328AT	8004
	1750	2100	366AT	9275	366AT	9275
	1150	2600	368AT	11928	368AT	11928
	850	1700	407AT	14561	368AT	14561
	650	1600	504AT	17483	409AT	17483
	500	1500	504AT	21153	504AT	21153
75	2500	2700	506AT	25016	506AT	25016
	1750	2100	508AT	30389	508AT	30389
	1150	2600	366AT	10922	366AT	10922
	850	1700	L407AT	14048	407AT	14048
	650	1600	L409AT	16731	409AT	16731
	500	1500	504AT	20420	504AT	20420
100	2500	2700	506AT	24393	506AT	24393
	1750	2100	508AT	28740	508AT	28740
	1150	2600	368AT	13737	368AT	13737
	850	1700	L407AT	17199	407AT	17199
	650	1600	L409AT	20183	409AT	20183
	500	1500	506AT	24470	506AT	24470
125	2500	2700	508AT	29016	508AT	29016
	1750	2100	4352	35199	4355	35199
	1150	2600	368AT	13737	368AT	13737
	850	1700	L407AT	17199	407AT	17199
	650	1600	L409AT	20183	409AT	20183
	500	1500	506AT	24470	506AT	24470
150	2500	2700	508AT	29016	508AT	29016
	1750	2100	4352	35199	4355	35199
	1150	2600	368AT	13737	368AT	13737
	850	1700	L407AT	17199	407AT	17199
	650	1600	L409AT	20183	409AT	20183
	500	1500	506AT	24470	506AT	24470
200	2500	2700	508AT	29016	508AT	29016
	1750	2100	4352	35199	4355	35199
	1150	2600	368AT	13737	368AT	13737
	850	1700	L407AT	17199	407AT	17199
	650	1600	L409AT	20183	409AT	20183
	500	1500	506AT	24470	506AT	24470
250	2500	2700	508AT	29016	508AT	29016
	1750	2100	4352	35199	4355	35199
	1150	2600	368AT	13737	368AT	13737
	850	1700	L407AT	17199	407AT	17199
	650	1600	L409AT	20183	409AT	20183
	500	1500	506AT	24470	506AT	24470

\* Ratings normally carried in stock.

**IMPORTANT:** See Page 24 for explanatory notes and publications.

Prices and data subject to change without notice

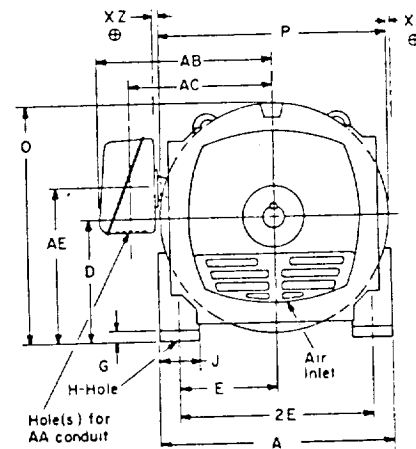
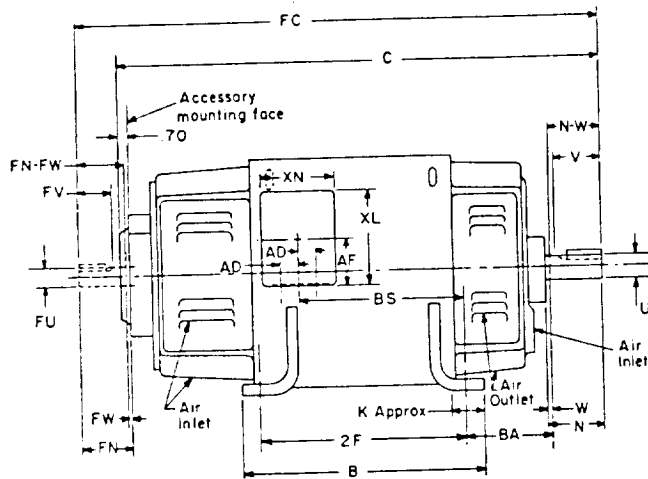
GENERAL ELECTRIC

FOOT MOUNTED

**Industrial KINAMATIC®**

Dripproof Fully Guarded,\* Splashproof†, and Totally Enclosed Nonventilated  
Type CD, Frames 365AT to L508AT

**DIMENSIONS—For ESTIMATING ONLY unless endorsed for construction.**



Frame	Approx Net Wt in Lb	Approx Wt of Arm. in Lb-ft	DRIVE END			COMMUTATOR END			Dimensions in Inches										
			KEY			KEY			A Max	B Max	C	D †	E	2F	G	H	J	K	N
			Width	Thick	Length -0.03	Width	Thick	Length -0.03											
365AT	750	15.61	.625	.625	3.50	.50	.50	3.00	17.92	14.16	33.70	9.00	7.00	12.25	.74	.8125	3.26	2.31	4.92
366AT	860	18.27	.625	.625	3.50	.50	.50	3.00	17.92	15.90	35.90	9.00	7.00	14.00	.74	.8125	3.26	2.31	4.92
368AT	1020	22.21	.625	.625	3.50	.50	.50	3.00	17.92	19.90	38.90	9.00	7.00	18.00	.74	.8125	3.26	2.31	4.92
407AT	1300	35.47	.625	.625	4.00	.625	.625	3.50	20.00	20.16	40.12	10.00	8.00	18.00	.86	.9375	4.00	2.38	5.42
L407AT	1350	35.54	.625	.625	4.00	.625	.625	3.50	20.00	20.16	43.52	10.00	8.00	18.00	.86	.9375	4.00	2.38	5.42
409AT	1600	43.81	.625	.625	4.00	.625	.625	3.50	20.00	24.16	44.62	10.00	8.00	22.00	.86	.9375	4.00	2.38	5.42
L409AT	1650	43.88	.625	.625	4.00	.625	.625	3.50	20.00	24.16	48.02	10.00	8.00	22.00	.86	.9375	4.00	2.38	5.42
504AT	1900	79.10	.75	.75	5.25	.75	.75	4.50	24.92	18.96	45.74	12.50	10.00	16.00	1.11	1.1875	4.50	3.00	6.67
L504AT	2070	79.15	.75	.75	5.25	.75	.75	4.50	24.92	18.96	47.50	12.50	10.00	16.00	1.11	1.1875	4.50	3.00	6.67
506AT	2290	98.76	.75	.75	5.25	.75	.75	4.50	24.92	22.96	49.74	12.50	10.00	20.00	1.11	1.1875	4.50	3.00	6.67
L506AT	2440	98.81	.75	.75	5.25	.75	.75	4.50	24.92	22.96	51.50	12.50	10.00	20.00	1.11	1.1875	4.50	3.00	6.67
508AT	2810	121.87	.75	.75	5.25	.75	.75	4.50	24.92	27.96	54.74	12.50	10.00	25.00	1.11	1.1875	4.50	3.00	6.67
L508AT	2970	122.92	.75	.75	5.25	.75	.75	4.50	24.92	27.96	56.50	12.50	10.00	25.00	1.11	1.1875	4.50	3.00	6.67

Frame	Dimensions in Inches													BS				XZ ⊕
	O	P	U □	V Δ	W	N-W	BA	FC	FN	FU □	FV Δ	FW	FN-FW	AA = 3"	AA = 4"	AA = (2)4"	AA = Blank	
365AT	17.91	17.90	2.375	4.50	.17	4.75	5.875	37.45	4.45	2.125	4.00	.20	4.25	9.02	9.02	6.64	.....	0.25
366AT	17.91	17.90	2.375	4.50	.17	4.75	5.875	39.65	4.45	2.125	4.00	.20	4.25	11.22	11.22	8.84	.....	.25
368AT	17.91	17.90	2.375	4.50	.17	4.75	5.875	42.65	4.45	2.125	4.00	.20	4.25	14.22	14.22	11.84	.....	.25
407AT	20.15	20.38	2.625	5.00	.17	5.25	6.625	44.37	4.95	2.375	4.50	.20	4.75	15.18	15.18	12.80	12.80	...
L407AT	20.15	20.38	2.625	5.00	.17	5.25	6.625	47.77	4.95	2.375	4.50	.20	4.75	15.18	15.18	12.80	12.80	...
409AT	20.15	20.38	2.625	5.00	.17	5.25	6.625	48.87	4.95	2.375	4.50	.20	4.75	19.68	19.68	17.30	17.30	...
L409AT	20.15	20.38	2.625	5.00	.17	5.25	6.625	52.27	4.95	2.375	4.50	.20	4.75	19.68	19.68	17.30	17.30	...
504AT	25.15	25.38	3.250	6.25	.17	6.50	8.50	50.99	5.95	2.875	5.50	.20	5.75	.....	13.26	10.88	10.88	...
L504AT	25.15	25.38	3.250	6.25	.17	6.50	8.50	52.75	5.95	2.875	5.50	.20	5.75	.....	.....	.....	11.16	...
506AT	25.15	25.38	3.250	6.25	.17	6.50	8.50	54.99	5.95	2.875	5.50	.20	5.75	.....	17.26	14.88	14.88	...
L506AT	25.15	25.38	3.250	6.25	.17	6.50	8.50	56.75	5.95	2.875	5.50	.20	5.75	.....	.....	19.88	19.88	...
508AT	25.15	25.38	3.250	6.25	.17	6.50	8.50	59.99	5.95	2.875	5.50	.20	5.75	.....	22.26	.....	.....	...
L508AT	25.15	25.38	3.250	6.25	.17	6.50	8.50	61.75	5.95	2.875	5.50	.20	5.75	.....	.....	.....	20.16	...

\* Dripproof fully guarded machines can be used for wall or ceiling mounting. Assembly modifications must be made to maintain proper enclosure.

† Dimension D will not be exceeded. When exact dimension is required, shims up to 0.06 inch may be necessary.

⊕ Splashproof machines will have additional covers which may increase the over-all width at the commutator end and drive-end side cover openings.

Dimension "V" represents the minimum length of shaft available for hubs.

□ Shaft diameters will come within the limits of +0.0000 inch -0.0010 inch. Runout of shaft diameters shall not exceed 0.003-inch total indicator reading.

The standard single-shaft machine has the commutator-end bearing bracket and shaft prepared accept accessories. For additional information see 36C697103AA.

Commutator-end shaft extension is furnished only when specifically ordered and is prepared for accessory drive.

Conduit box will be assembled on the right-hand side facing the commutator end for motors and on the left-hand side facing the commutator end for generators. Conduit box will be assembled on the opposite side of frame if so specified. Conduit box may be turned so that entrance may be made upward, downward, from commutator end or drive end. Dimensions pertaining to conduit boxes vary according to rating. Refer to company for dimensions. For shipping weight add 15 percent to net weights.

Prices and data subject to change without notice

From 36C697106AA

GENERAL ELECTRIC

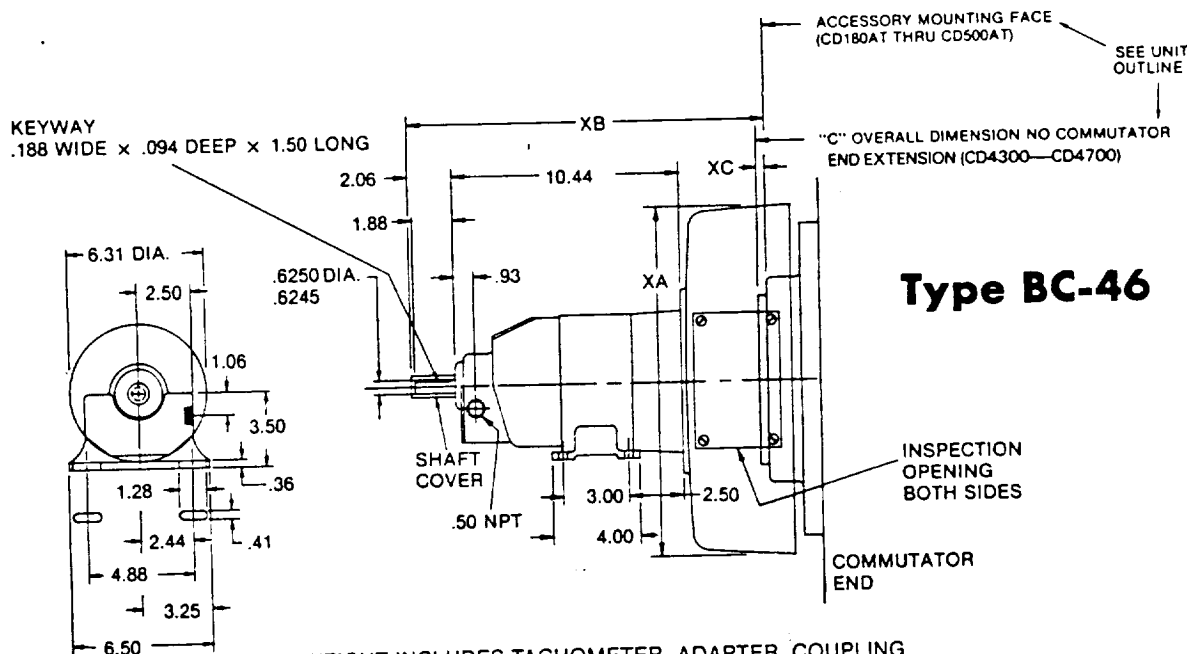
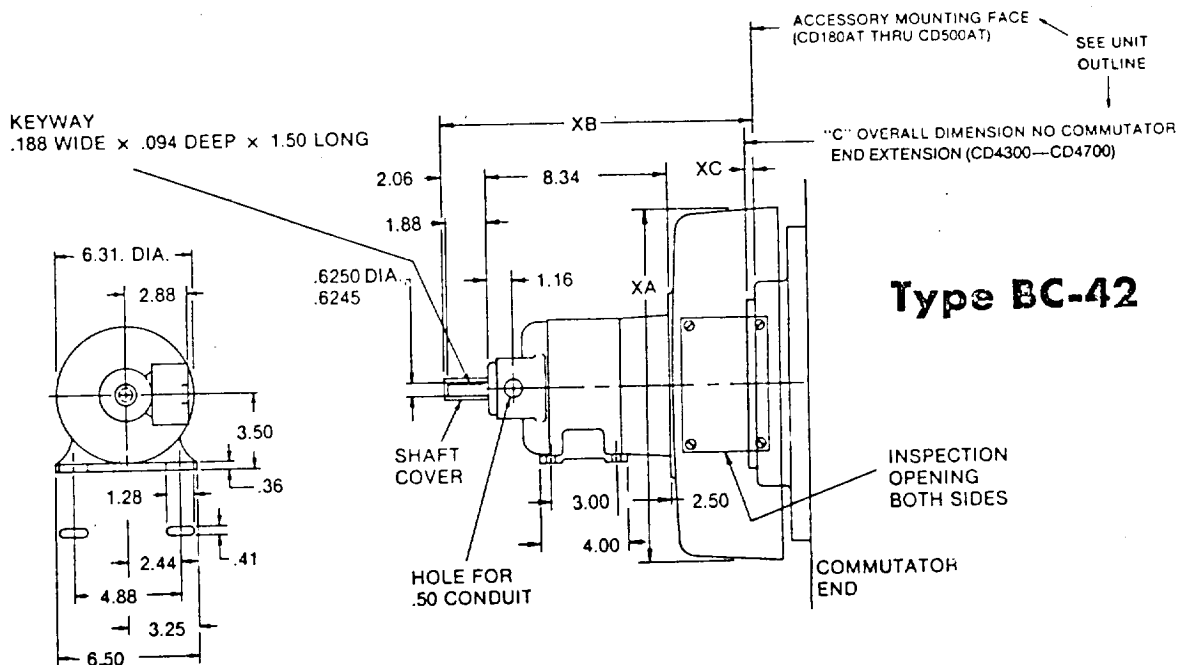
# Type BC Tachometer Generator

**KINAMATIC® Motors**

Type CD, Frames 180AT through 4700

USE 368 AT -

**DIMENSIONS—For ESTIMATING ONLY unless endorsed for construction**



WEIGHT INCLUDES TACHOMETER, ADAPTER, COUPLING

FRAME	BC-42 APPROX. NET WT.	BC-46 APPROX. NET WT.	XA	BC42 XB	BC46 XB	XC
CD180AT	35 LB.	45 LB.	8.88 DIA.	14.65	16.75	OMIT
CD210AT THRU CD500 AT	35 LB.	45 LB.	8.88 SQUARE	14.65	16.75	OMIT
CD4300 & CD4400	45 LB.	55 LB.	13.00 DIA.	14.40	16.50	.40
CD4500—CD4700	55 LB.	65 LB.	16.25 DIA.	14.40	16.50	.40

Prices and data subject to change without notice

From 36C695002BK, 36C695001BC

GENERAL ELECTRIC

OUTLINE  
DIMENSIONS

## Accessories and Modifications

### Dripproof Fully Guarded and Totally Enclosed **KINAMATIC®** Motors

USE 368 AT

The list price additions for all modifications and accessories apply to the basic list price of both dripproof, fully guarded and totally enclosed Kinamatic motors unless otherwise noted.

ITEM	DESCRIPTION					
<b>6</b> Cont'd	<p>TENV and TEAO motors for applications in ambients containing chips, nonabrasive, nonconducting and non-explosive dusts and coolants.</p> <p><b>WARNING:</b> Brake performance and features must be carefully matched to the requirements of the application. Consideration must be given to torque requirements, especially where an overhauling condition exists, as well as thermal capacity, ambient temperature, atmospheric explosion hazards, type of enclosure, power supply characteristics and other unusual conditions. Improper selection or installation of a brake and/or lack of maintenance may cause brake failure which can result in damage to property and/or injury to personnel. If injury to personnel could be caused by brake failure, proper safeguards must be provided to insure safety of personnel.</p> <p>A space heater should be specified whenever a heater is required on the motor. For brake space heater add <b>\$122, GO-2A</b>.</p> <p>The standard coil voltages are 230/460 volts a-c single-phase. Brakes through 105 lb. ft. are rated 50/60 hertz. On larger brakes, add <b>\$113 GO-2A</b> for 50 hertz. D-c coil volts are 120 or 240.</p> <p><b>DPFG, TENV OR TEUC MOTORS <math>\phi</math></b></p>					
		Mounted Brake				
		A-c Coil		D-c Coil		
		Standard Enclosure	Dust Tight-Waterproof	Standard Enclosure	Dust Tight-Waterproof	
		List Price Addition				
ft. lb. $\Delta$	Available Frames					
1.5	180AT	\$558	\$652	\$666	\$873	
3	180AT	606	710	720	1025	
6	180AT	606	755	788	1035	
9	180AT	630	755	810	1115	
10	180AT-218AT	777	810	810	1160	
15	180AT-2110AT	833	923	990	1238	
25	218AT-259AT	945	1260	1283	1553	
35	218AT-288AT	1113	1395	1373	1733	
50	219AT-328AT	1283	1598	1508	1958	
75	2110AT-328AT	1733	2115	1800	2325	
105	259AT-328AT	2250	2700	2475	3195	
125	287AT-409AT	3038	3600	3488	4230	
175	288AT-409AT	4298	4973	4523	5513	
230	328AT-409AT	4838	5558	5423	6615	
330	365AT-409AT	6008	6323	7155	7853	

$$\Delta \text{ Ft lb torque} = \frac{\text{Hp} \times 5252}{\text{Motor base speed (rpm)}}$$

$\phi$  Refer to company on use of TEFC with brakes. Also see enclosure modification for TEAO (totally enclosed, air-over frame construction).

FOR SPECIAL BRAKES WITH A THROUGH SHAFT and provision for mounting a tachometer or speed limit switch on DPFG or TENV motors, add to the brake price as follows: (Refer to company on waterproof motors).

Brake Torque Lb Ft	Price Adder
1.5—6	\$158
9—15	221
25—50	284
75—105	696
125—330 and above	1514

#### SHOE-TYPE

Shoe-type brakes are available for floor mounting only. Refer to the applicable handbook section for brake selection, description and prices and add for standard, double, shaft extension on the motor.

Prices and data subject to change without notice

# APPENDIX C

## Anchor Calculations

$$\text{Angle of repose} = 30^\circ$$

$$V = \frac{1}{3}\pi r^2 h$$

$$r = h \tan 30$$

$$\text{Density} = 3000 \text{ kg/m}^3 = 187.43 \text{ lb/ft}^3$$

$$g_{\text{moon}} = (9.81)(\frac{1}{6}) = 1.635 \text{ m/sec}^2$$

$$1 \text{ lb} = 4.448 \text{ N}$$

1 meter auger

$$h = 1 \text{ m}$$

$$r = 1(\tan 30) = .577$$

$$V = (\frac{1}{3})\pi (.577)^2 (1) = .349 \text{ m}^3$$

$$\text{Mass} = (3000)(.349) = 1047.0 \text{ kg}$$

$$\begin{aligned}\text{Force} &= (1047)(1.635) = 1711.945 \text{ N} \\ &= 384.94 \text{ lbs}\end{aligned}$$

2 meter auger

$$h = 2 \text{ m}$$

$$r = 1.1547 \text{ m}$$

$$V = 2.793 \text{ m}^3$$

$$\text{Mass} = 8377.58 \text{ kg}$$

$$\begin{aligned}F &= 13,823.01 \text{ N} \\ &= 3107.54 \text{ lb}\end{aligned}$$

1.5 meter auger

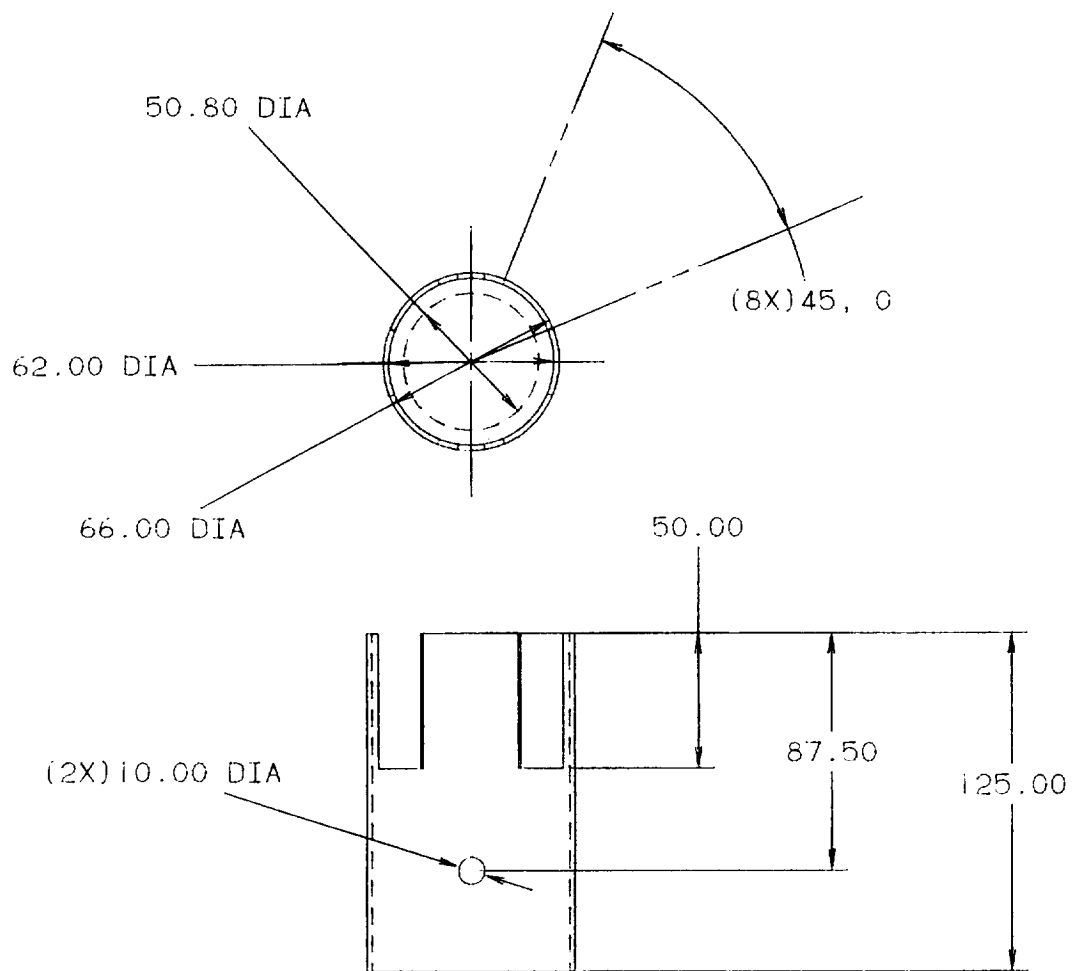
$$h = 1.5$$

$$r = .866$$

$$V = 1.178 \text{ m}^3$$

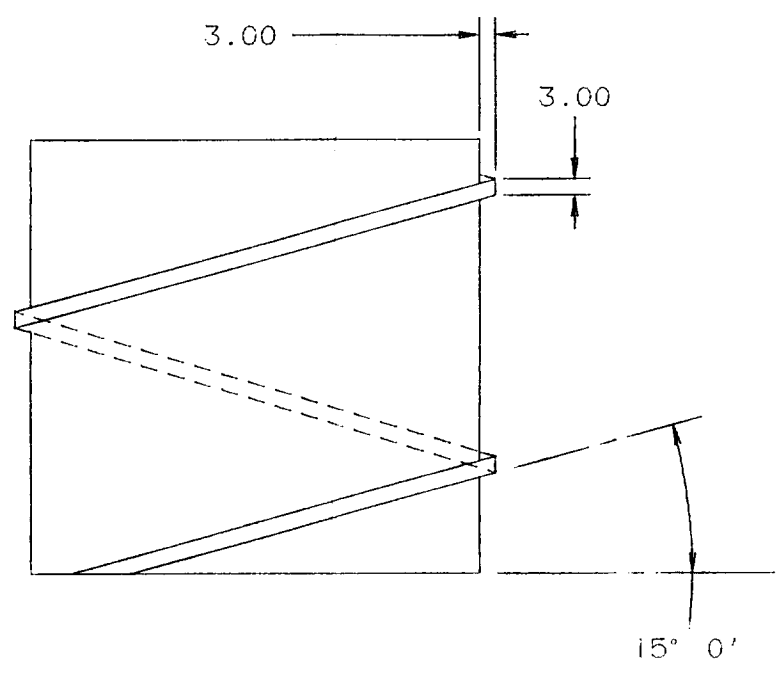
$$\text{Mass} = 3,534.3 \text{ kg}$$

$$\begin{aligned}F &= 5778.57 \text{ N} \\ &= 1299.07 \text{ lb}\end{aligned}$$



SEE DETAIL A-A

**FOLDOUT FRAME** 1



DETAIL A-A  
(AUGERS)

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: ANCHOR AUGER	
DESIGN:	DATE 3/10/86
CHECK:	DATE
DRWG NO.	SCL

## APPENDIX D

### Bit-Generated Heat Transfer Considerations:

$T$  = torque at bit/rock interface [ft-lb<sub>f</sub>]

$P$  = downward force [lb<sub>f</sub>]

$r$  = mean radius [inches]

$f$  = coeff. of friction

HP = power [hp, B/hr]

$N$  = speed [rpm] = 210 rpm

$W_{\max}$  = 2000 lb<sub>f</sub>

$r$  = 1.476 in

$f$  = 0.5

$T_{\max} = f \cdot W_{\max} \cdot r$

$= 0.5 \cdot 2000 \cdot 1.476 = 1476.0 \text{ in-lb}_f$

$T_{\max} = 123 \text{ ft-lb}_f$

$HP = \frac{T \cdot N}{5252} = \frac{123 \cdot 210}{5252} = 4.92 \text{ hp}$

$HP = \frac{2 \cdot \pi \cdot 123 \cdot 210}{12.97} = 12513 \text{ B/hr}$

$HP_{\text{chips}} = 10010 \text{ B/hr}$

$HP_{\text{cond.}} = 2503 \text{ B/hr}$

$k_{\text{steel}} = 48.46 \text{ B/hr-ft-}^{\circ}\text{F} \text{ (@ } 1000^{\circ}\text{F)}$

$k_{2024-T4} = 277 \text{ B/hr-ft-}^{\circ}\text{F}$

$A$  = area

$A_{\text{bit}} = 0.038 \text{ ft}^2$

$A_{\text{shaft}} = 0.012 \text{ ft}^2$

$L$  = length

$L_{\text{bit}} = 0.0328 \text{ ft}$

$Q = 2503 \text{ B/hr}$

Appendix D: (cont'd)

$$T_{\text{bit,max}} = 500^{\circ}\text{F}$$

$$Q = -[k \cdot A \cdot (t_B - t_L)] / L$$

$$= 2503 = -[48.46 \cdot 0.038 \cdot (500 - t_L)] / 0.0328$$

$$t_{0.0328} = 455^{\circ}\text{F}$$

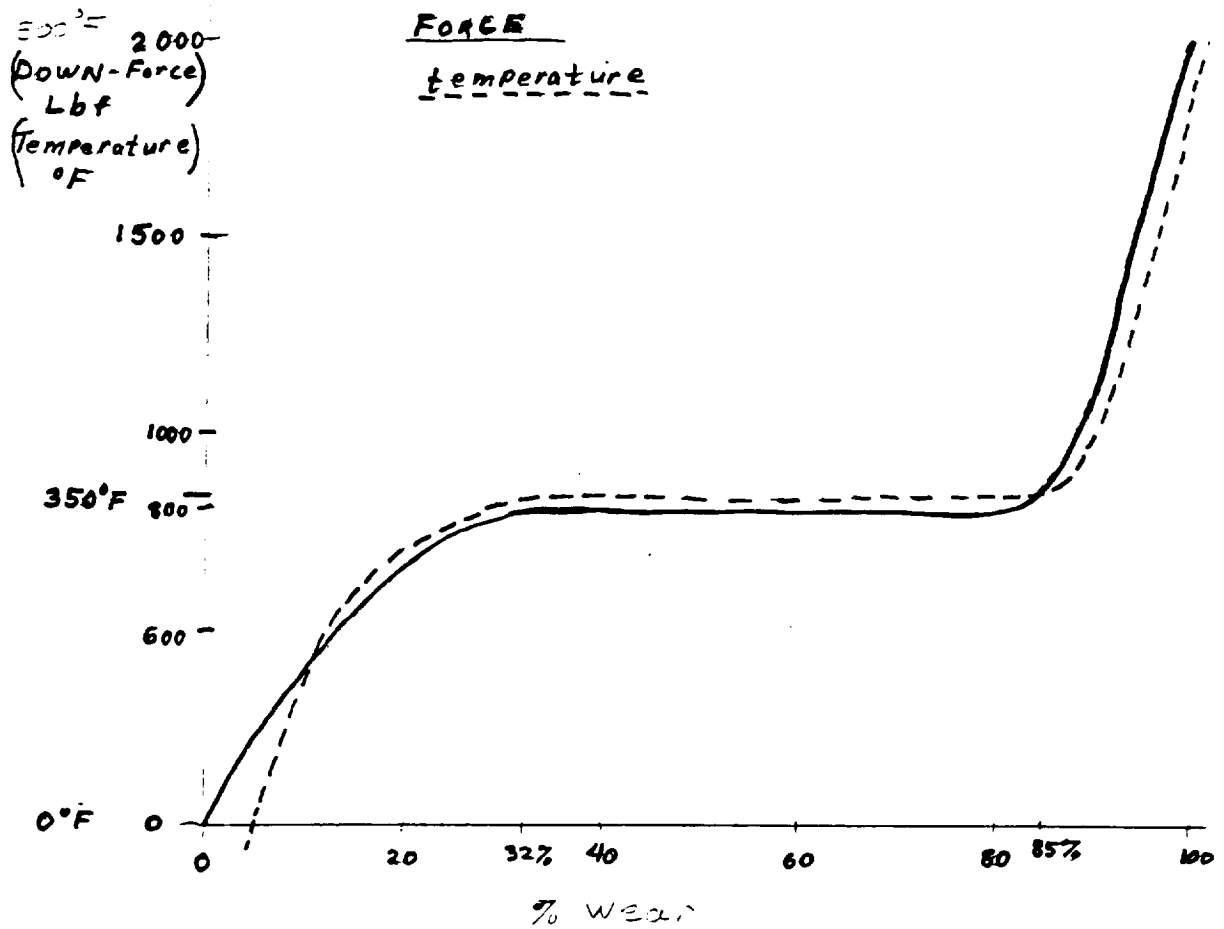
$$2503 = -[277 \cdot 0.012 \cdot (500 - 70)] / L_{70^{\circ}}$$

$$L_{70^{\circ}} = 0.57 \text{ ft}$$

Therefore, the temperature will be approx.  $70^{\circ}\text{F}$ , and  $1/3$  meter from bit/auger interface.

Appendix D: (cont'd)

Plot of  
Down-Force vs. % Wear and Temperature vs. % Wear



## APPENDIX E

### Auger Length Calculation

A 2 meter run is desired.

The volume of chips swept out is:

$$(PI/4)*((90)^2-(60)^2)*2000 = 7068583.47 \text{ mm}^3$$

The volume in the hole not occupied by the auger shaft must equal the chips' volume.

$$V = (PI/4)*((90)^2-(83)^2)*L-(3L*3*3)/(\sin 15^\circ) \text{ mm}^3$$

$$V = 846.8 L$$

$$846.8 L = 7068583.47$$

$$L = 8.3 \text{ meters}$$

An augered shaft length of 8m may be used without clogging of chips, because the standard drill string shaft is smaller in diameter than the auger.

### Drill String Shaft Strength Calculations:

Max. torque:  $T = 125.7 \text{ ft-lb}_f$

Max. compression:  $P = -2000 \text{ lb}_f$

Length:  $L$

Outer radius:  $R$

Inner radius:  $r$

Area:  $A$

Material Properties: Aluminum ASTM 2024-T4,

Yield Str.(tens./compr.) = 48000 psi

Yield Str.(shear) = 25000 psi

$G = 4.0E06 \text{ psi}$

$E = 10.6E06 \text{ psi}$

Appendix E: (cont'd)

$$J = 2*PI(R-r)((R+r)/2)^3$$

$$I = J/2$$

$$\sigma_x = P/A \quad , \quad \tau_{xy} = T*R/J \quad , \quad \sigma_0 = \sigma_x/2 \quad (\sigma_y = 0)$$

$$\sigma_{max} = \sigma_0 + (\sigma_x^2 + \tau_{xy}^2)^{1/2}$$

$$\sigma_{min} = \sigma_0 - (\sigma_x^2 + \tau_{xy}^2)^{1/2}$$

For R = 39mm, r = 35mm, T = 125.7 ft-lb<sub>f</sub> , P = -2000 lb<sub>f</sub> :

$$J = 2*PI(39-35)((39-35)/2)^3$$

$$= 1,273,048.72 \text{ mm}^4 \quad (1 \text{ in}/25.4\text{mm})^4$$

$$J = 3.0508 \text{ in}^4$$

$$A = PI(R^2-r^2) = PI(39^2-35^2)$$

$$= 929.91 \text{ mm}^2 = 1.4414 \text{ in}^2$$

$$\sigma_x = -2000/1.4414$$

$$= 1.3876E03 \text{ psi} \quad , \quad \text{compression}$$

$$\sigma_0 = -693.79 \text{ psi}$$

$$\tau_{xy} = 125.7*(39*12)*(1/25.4)/3.0585$$

$$= 757.25 \text{ psi}$$

$$\sigma_{max} = -693.79 - [(1.3876E03)^2 + (757.25)^2]^{1/2}$$

$$\sigma_{max} = -2274.6 \text{ psi}$$

$$\sigma_y/\sigma_{max} = n_c = 21.1$$

$$\tau_{max} = (\sigma_x^2 + \tau_{xy}^2)^{1/2}$$

$$= 1580.0 \text{ psi}$$

$$25000/1580.8 = n_s = 15.8$$

Appendix E: (cont'd)

Buckling:

Euler's Formula:

$$P_{cr} = C \cdot \pi^2 \cdot E / (l/k)^2$$

J.B. Johnson's Formula:

$$P_{cr} = A(a - b(l/k)^2) \quad , \quad a = S_Y$$

$$b = (S_Y / (2 \cdot \pi))^2 \cdot (1 / (C \cdot E))$$

$$(l/k)_{crit} = ((2 \cdot \pi^2 \cdot C \cdot E) / S_Y)^{1/2}$$

$$l/k = 4 \cdot 39.37 / (3.0585 / (2 \cdot 1.4414)) = 152.89$$

$$\begin{aligned} (l/k)_{crit} &= ((2 \cdot \pi^2 \cdot (10.6E06) \cdot 1.2) / 48000)^{1/2} \\ &= 72.3 \end{aligned}$$

$l/k > (l/k)_{crit}$  Use Euler formula for buckling.

$$P_{cr} = (1.2) \cdot \pi^2 \cdot (10.6E06) / (152.89)^2$$

$$\begin{aligned} n_b &= 5370.7 / 2000 \\ &= 2.69 \end{aligned}$$

For twist:

$$d\theta/dx = T/GJ \quad \text{Impose a maximum twist of 0.6 deg/meter.}$$

$$= (125.7 \cdot 12 / 3.0585 \cdot 4E06) \cdot (180 / \pi) \cdot 39.37$$

$$= 0.278 \text{ deg/meter}$$

$$\begin{aligned} n_t &= 0.6 / 0.278 \\ &= 2.16 \end{aligned}$$

Appendix E: (cont'd)

Connector Size Pin Calculation:

The maximum shear encountered by the pin occurs when the core is being broken. The strongest rock that is expected is granite. The tensile strength is 5500 psi.

The cross-sectional area of the core is:

$$\begin{aligned} &= \text{PI}/4 * (60)^2 / (25.4)^2 \\ &= 4.38 \text{ in}^2 \end{aligned}$$

$$\sigma_r = 5500 \text{ psi} , A = 4.38 \text{ in}^2$$

$$F = \sigma_r * A = 24104 \text{ lb}_f ; \text{ Choose } 24500 \text{ lb}_f \text{ to include drill string weight.}$$

For a pin made of AISI #201 <sup>3</sup>/<sub>4</sub> hard stainless steel:

$$S_y = 135 \text{ kpsi}$$

$$F = 24500 \text{ lb}_f$$

$$24500 / 135000 = A = \text{PI} * d^2 / 4$$

$$d = 0.48 \text{ in} * 25.4 = 12.2 \text{ mm diameter}$$

For a 17 mm diameter pin:

$$A = \text{PI}/4 * (17)^2 * (1/25.4)^2 = 0.352 \text{ in}^2$$

$$\sigma = 24500 / 0.352 = 69814 \text{ psi}$$

$$135000 / 69638 = 1.93 = \text{safety factor}$$

Therefore, a 17mm pin diameter is acceptable.

Appendix E: (cont'd)

Drill Bit Specifications:

Type : Diamond/matrix

Diamond size : 100/carat

Diamond density : 42/tooth

# of teeth: 12

Chip channels: 12, 2mm wide

Thread size: M60-6

Max. diameter : 90mm

Core diameter : 60mm

Tooth depth (vert.) : 20mm

Tooth length : 17.64mm

Cutting area : 4987mm<sup>2</sup>

Matrix material: Type 4 powdered metal (proprietary)

Bit holder mat'l: AISI 1040 CR steel

Auger Shaft Specifications:

Outside diameter: 89mm

Length: 4 meters

Auger leads: 3, 3mm x 3mm

Shaft dia. below auger leads: 83mm

Wall thickness: 3mm

Weight: 36.8 lb<sub>f</sub> (approx. earth wt., both sections included)

Max. flex rate: 0.6°/meter

Max. torque applied: 125.7 ft-lb<sub>f</sub>

Max. compression: 2000 lb<sub>f</sub>

Appendix E: (cont'd)

Max. tension: 24500lb<sub>f</sub>

Material: ASTM 2024-T4 (Aluminum)

Plating: Titanium nitrate

Required: 2 sections; the ends of the shafts are as specified in drawings.

Drill String Coupling Sleeve Specifications:

O.D.<sub>max</sub>: 87mm

I.D.<sub>min</sub>: 79mm

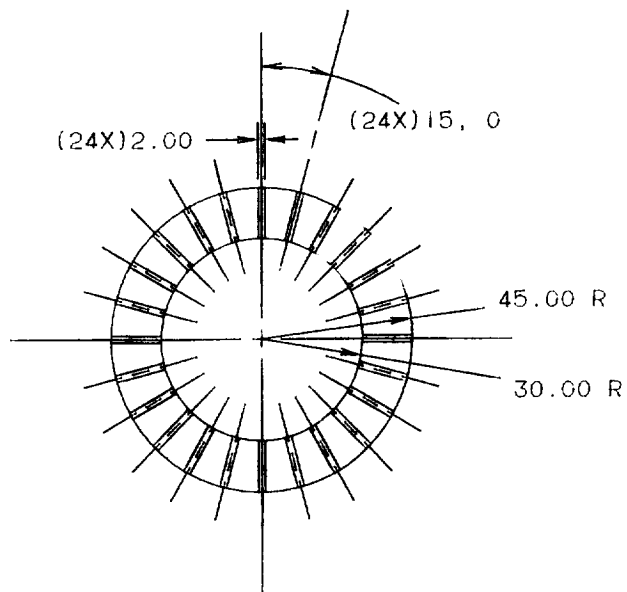
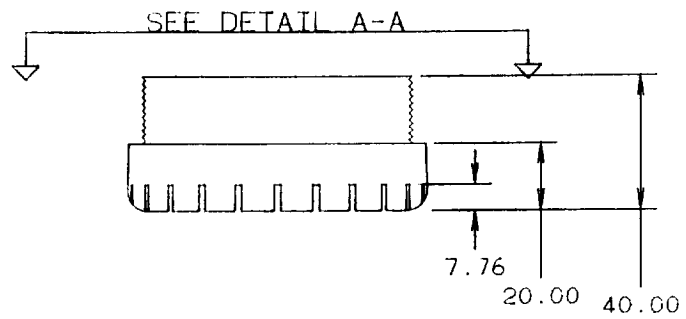
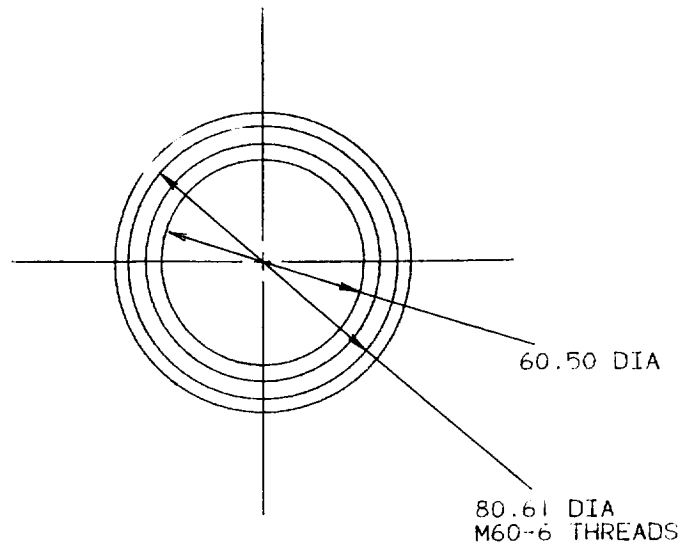
Wall thickness: 4mm

Length: 150mm

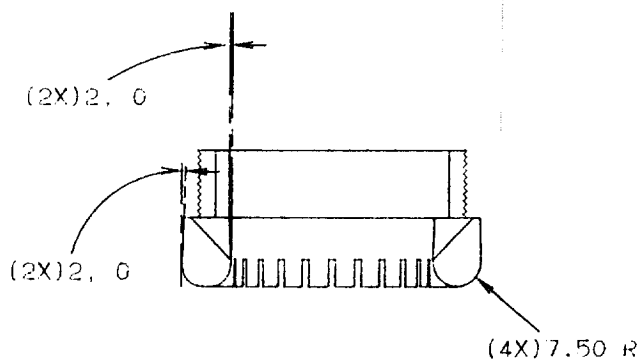
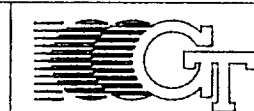
Material: ASTM 2024-T4 (Aluminum)

Weight: 2.8 lb<sub>f</sub>/section

Required: 23 pieces



FOLDOUT FRAME



DETAIL A-A

FOLDOUT FRAME 2

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: DRILL BIT	
DESIGN:	DATE 3/3/86
CHECK:	DATE
DRWG NO.	SCL 1/1

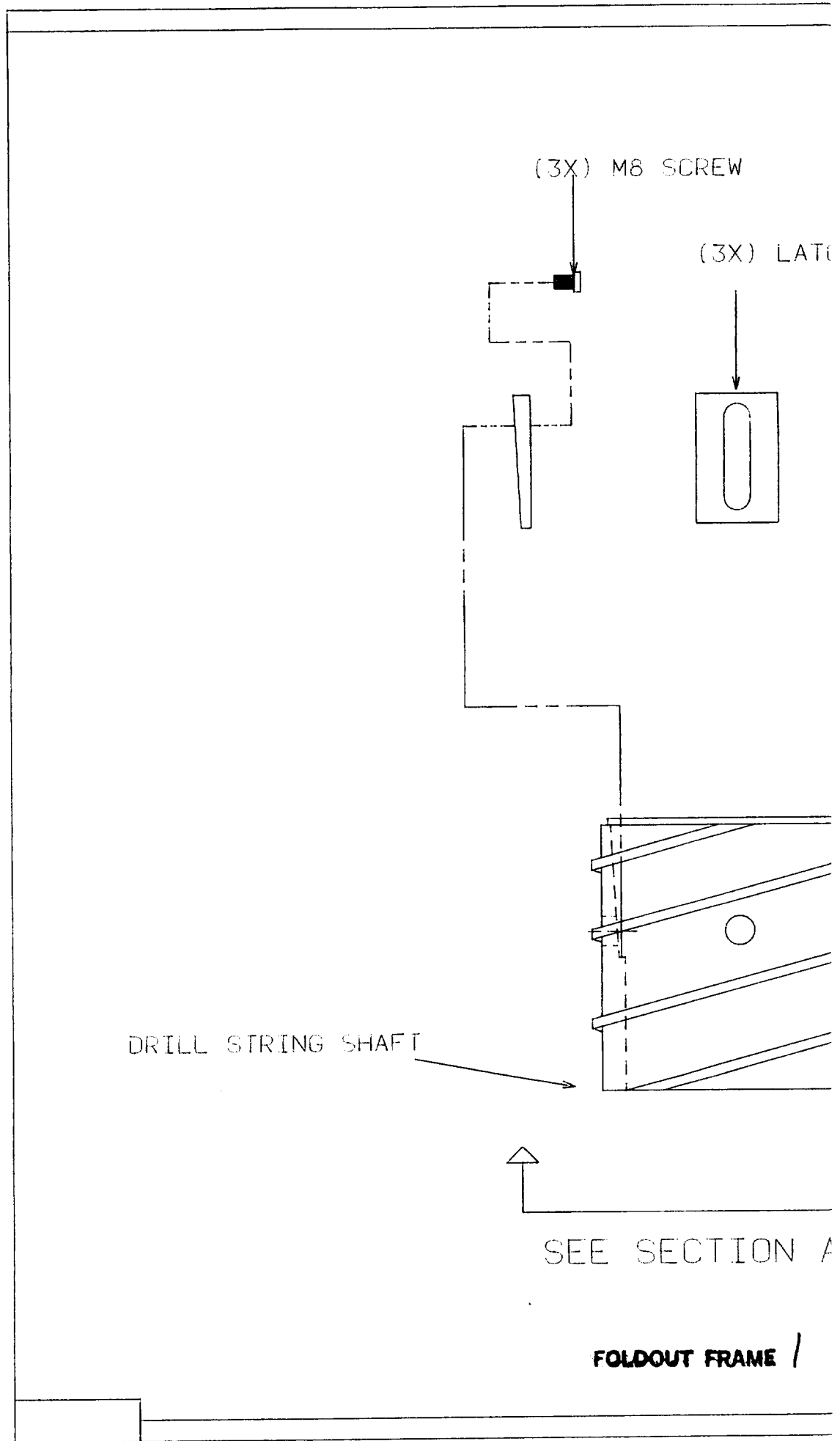
(3X) M8 SCREW

(3X) LAT

DRILL STRING SHAFT

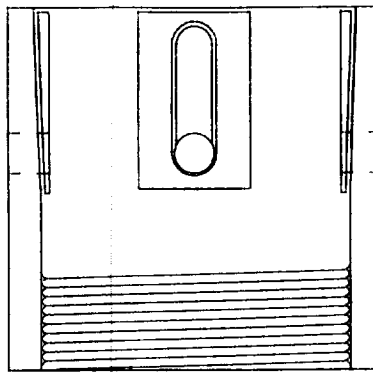
SEE SECTION A

FOLDOUT FRAME /

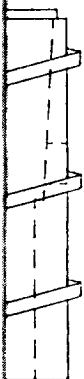




CHES



SECTION A-A

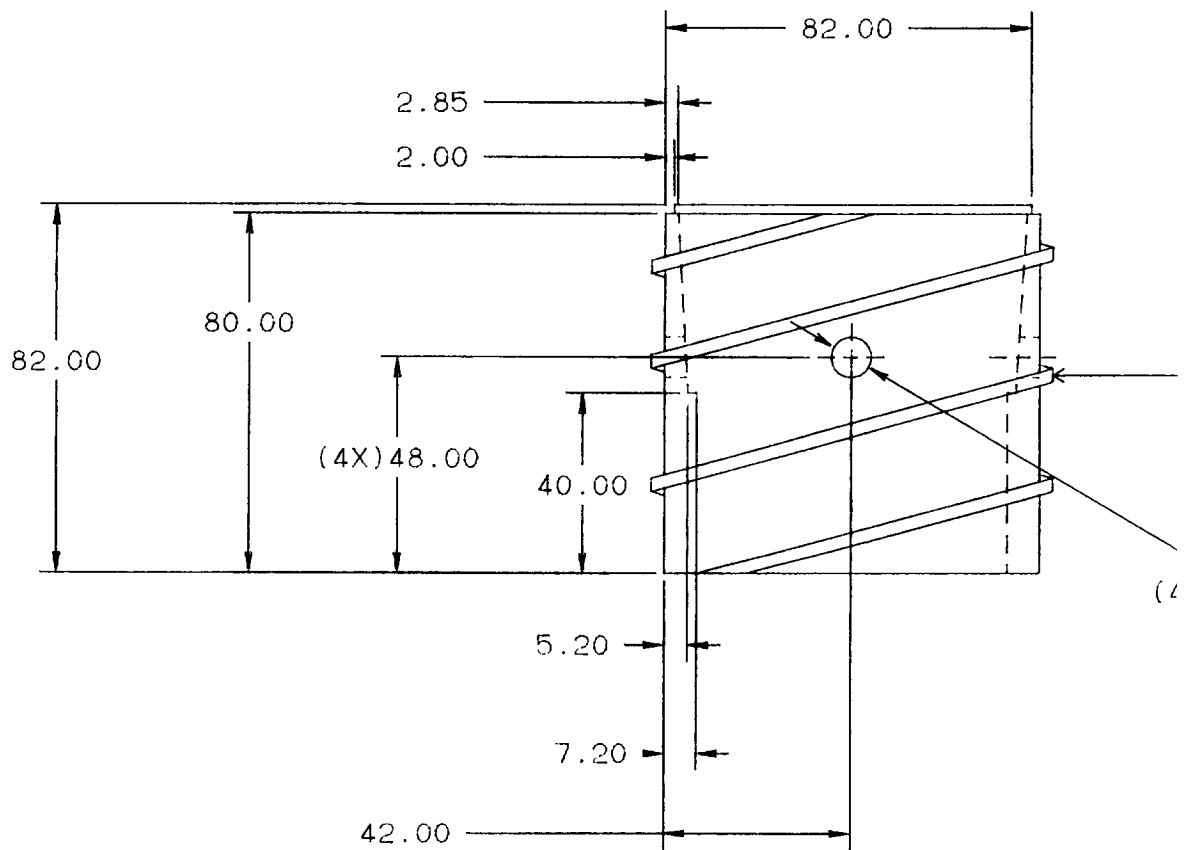


A-A

FOLDOUT FRAME 2

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: CORE RETRIEVAL	
DESIGN:	DATE
CHECK:	DATE
DRWG NO.	SCL

69.50 DIA



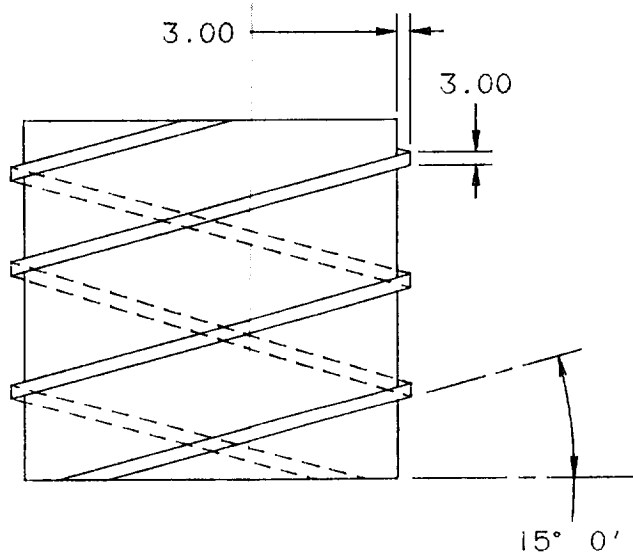
FOLDOUT FRAME 2



83.00 DIA

SEE DETAIL A

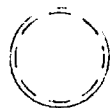
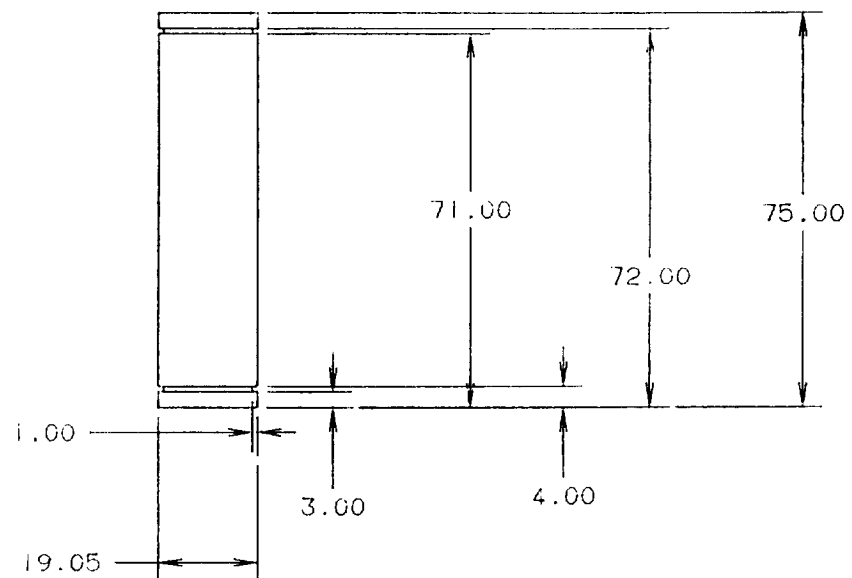
X) M9 HOLE



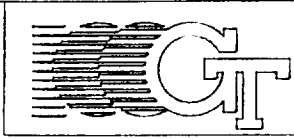
DETAIL A-A  
(AUGER)

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: AUGER/CORE REMOVAL	
DESIGN:	DATE 3/3/86
CHECK:	DATE
DRWG NO.	SCL

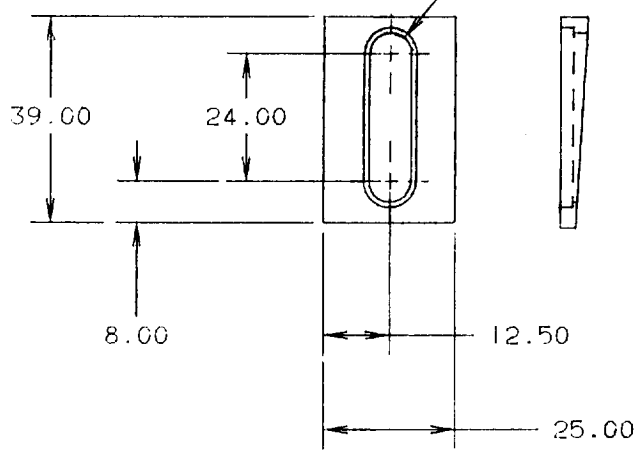
FOLDOUT FRAME 2



CONNECTOR PIN



(2X) 4.00 R THRU HOLE  
C'BORE 5.00 R 3 DP



# CORE LATCHES

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: LATCHES/PIN	
DESIGN:	DATE
CHECK:	DATE
DRWG NO.	SCL

1.50

12.70

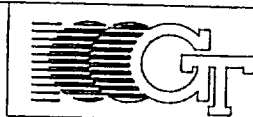
SEE DETAIL A-A

83.00 DIA

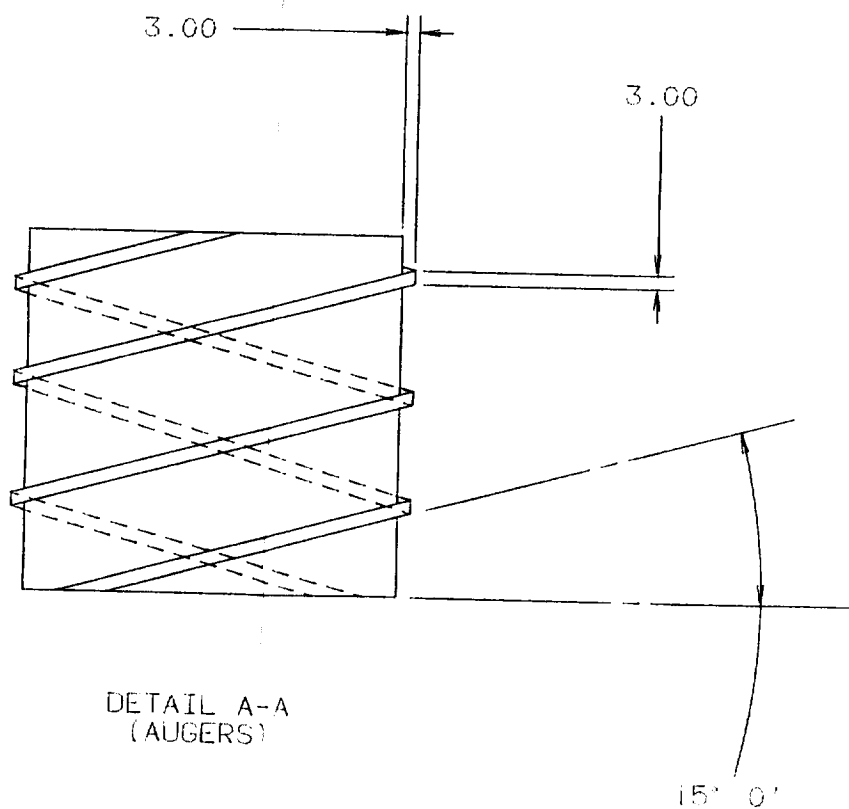
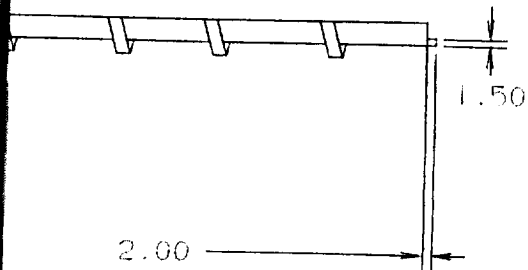
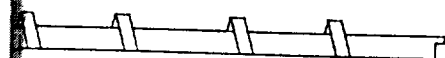
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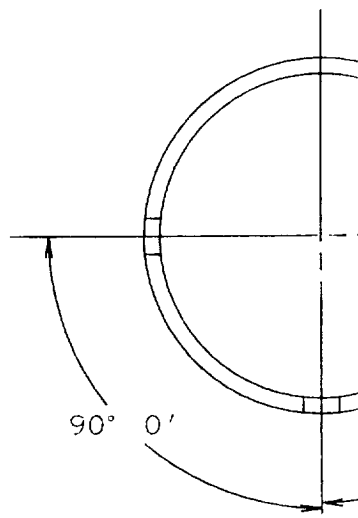
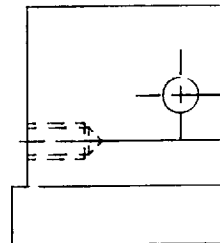
1 LENGTH OF SHAFT  
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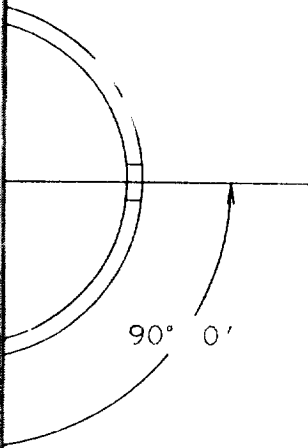
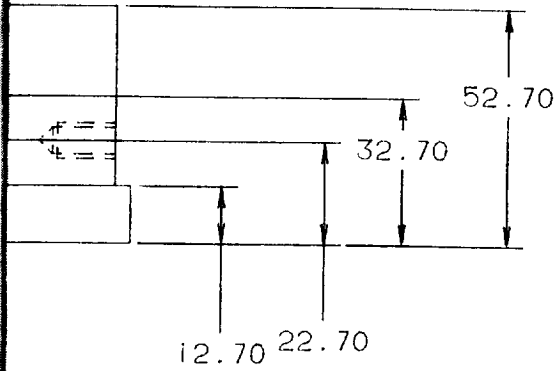
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COLLEGE OF ENGINEERING	
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CHECK:	DATE
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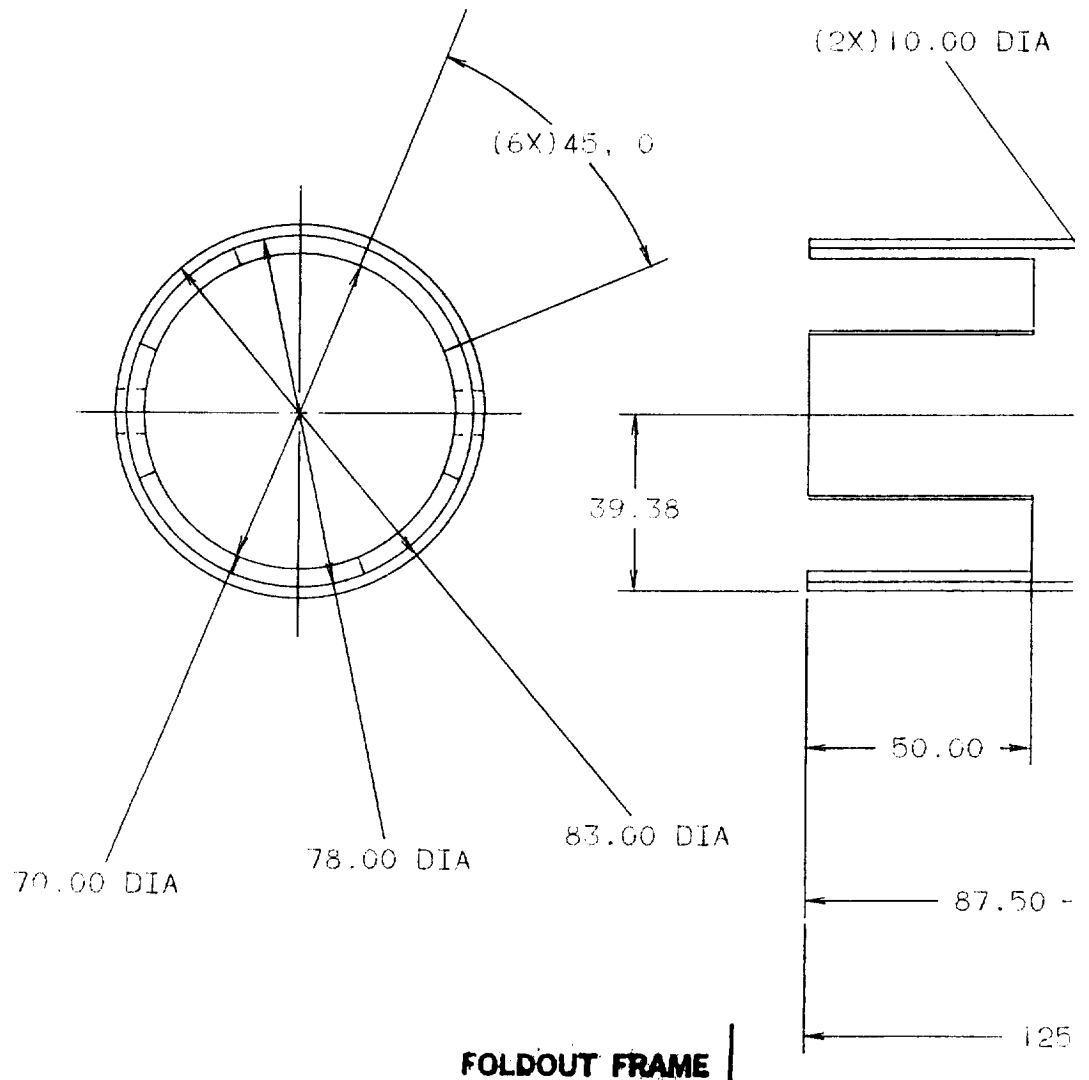
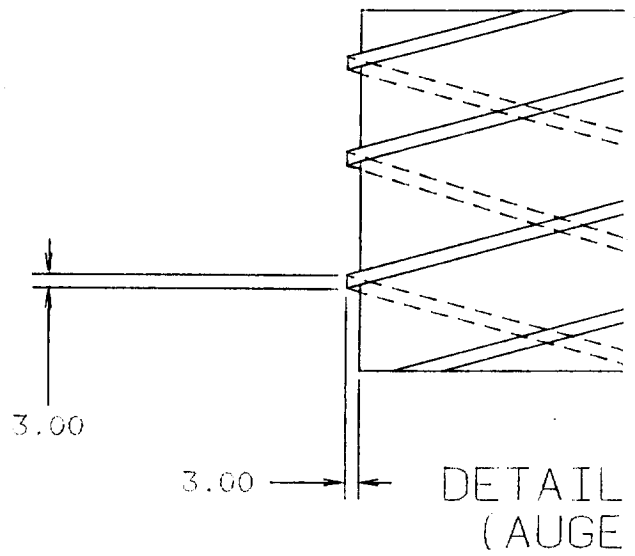


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FOLDOUT FRAME 2

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TITLE: CONNECTOR, AUGER	
DESIGN:	DATE 3/3/86
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|| LENGTH OF SHAFT  
IS 4 METERS

A-A  
(RS)

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(3X) 8.00 DIA

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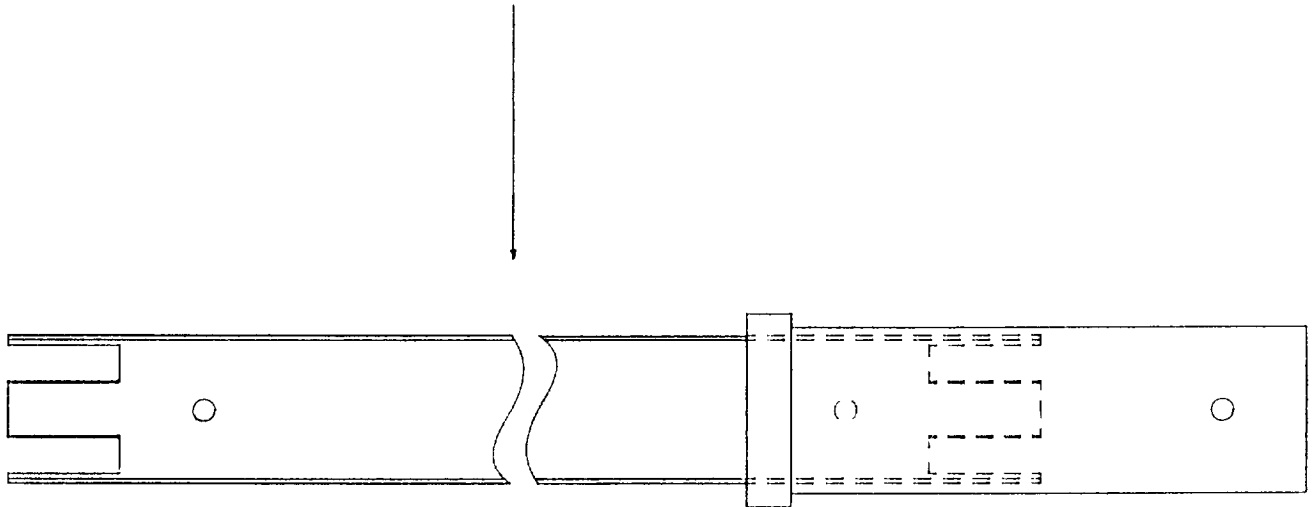
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SEE DETAIL A-A

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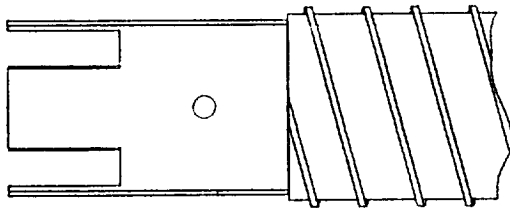
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DRILL STRING SHAFT



SLEEVE ASM

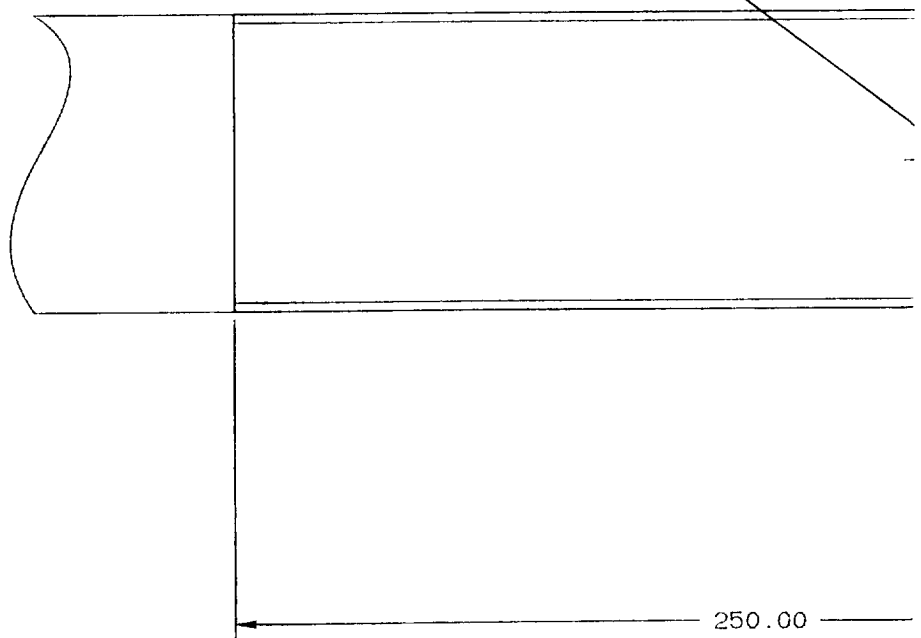
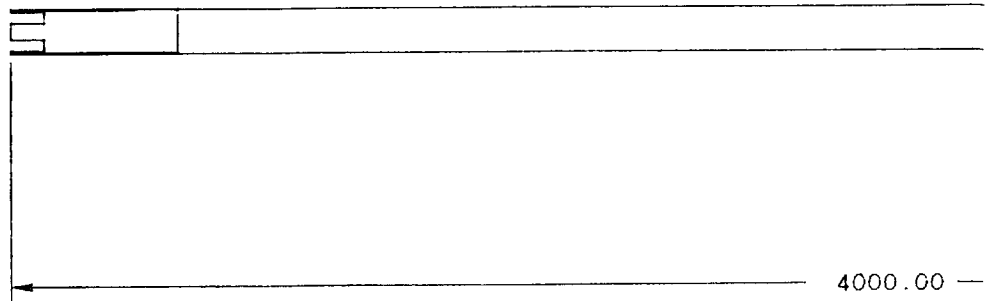
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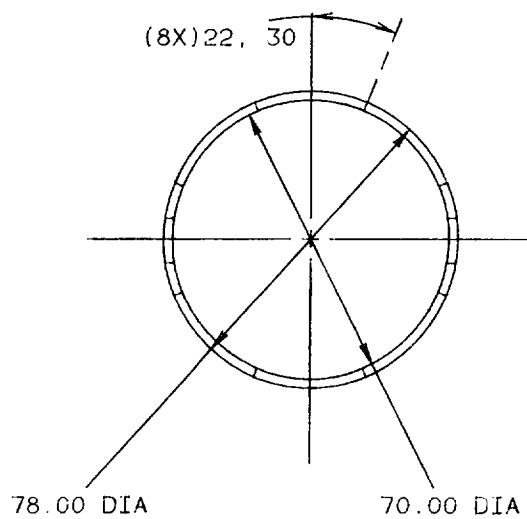
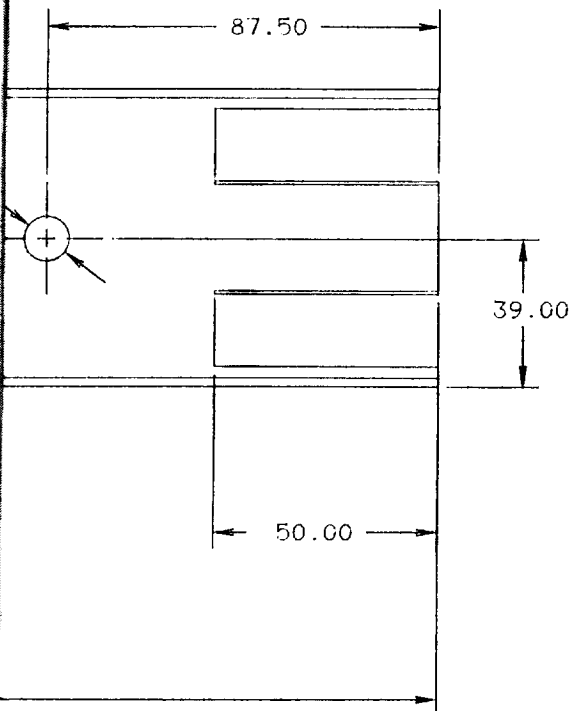
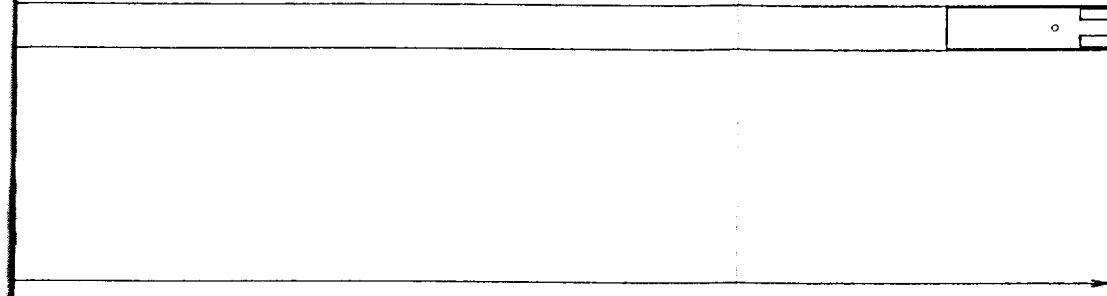
AUGER ASM

GEORGIA TECH	
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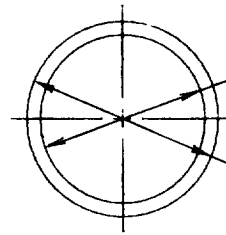


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COLLEGE OF ENGINEERING	
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DESIGN:	DATE 3/3/86
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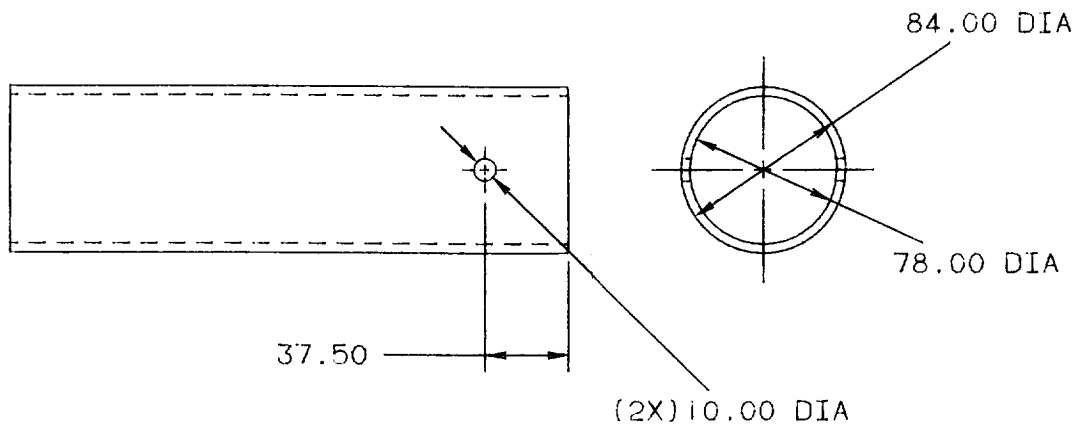
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88.00 DIA

COLLAR

COLLAR\_\_\_\_

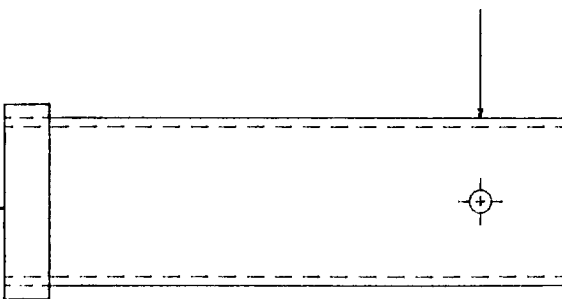
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SLEEVE

WELD CONNECTION

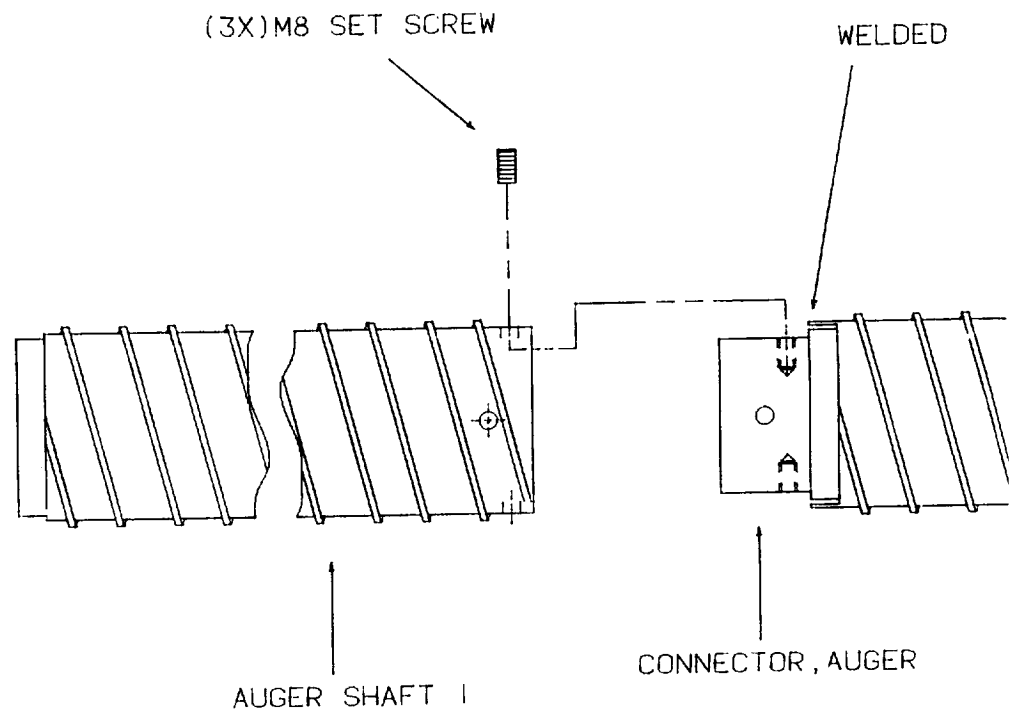
SLEEVE



SLEEVE ASM

FOLDOUT FRAME 2

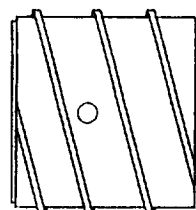
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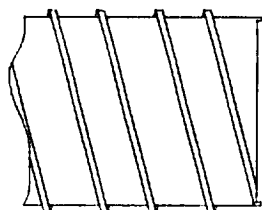
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DRILL BIT



CORE REMOVAL ASM



AUGER SHAFT 2

FOLDOUT FRAME 2

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: AUGER ASM	
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## APPENDIX F

### Heat Transfer

Assumptions :

- (1) Efficiencies of motors is 90%.
- (2) System must be designed for a combined motor output of 30 Hp.
- (3) Heat loss =  $\frac{P_{out}}{\eta} - P_{out}$
- (4) Conductivity of motor housing :  
 $k = 35 \text{ W/m-K}$
- (5) Operating time = 40 hrs =  $1.44 \times 10^6 \text{ sec}$
- (6) Initial temperature of system
- (7) System pressurized with hydrogen at 15 psi.

Area of 10 Hp motor :

$$L = 41 \text{ inches} = 1.04 \text{ m}$$

$$D = 18 \text{ inches} = 0.46 \text{ m}$$

Surface Area :

$$\begin{aligned} SA &= \pi DL + 2\pi r^2 = \pi [(0.46 \text{ m})(1.04 \text{ m}) + 2(0.23 \text{ m})^2] \\ &= 1.84 \text{ m}^2 \end{aligned}$$

## Appendix F: (cont'd)

Flow of hydrogen over motor will be assumed to be similar to that over a cylinder, where:

Properties of hydrogen at  $320^\circ\text{K}$ :

$$\alpha = 1.745 \times 10^{-4} \text{ m}^2/\text{s}$$

$$k = 0.192 \text{ W/m}\cdot\text{K}$$

$$\rho = 0.07717 \text{ kg/m}^3$$

$$\mu = 9.359 \times 10^{-6} \text{ N/m}^2$$

$$Pr = 0.7024$$

$$C_p = 14363 \text{ J/kg}\cdot\text{K}$$

Let velocity of hydrogen over motor =  $5 \text{ m/s}$ .

$$\begin{aligned} Re_D &= \frac{\rho D v}{\mu} = \frac{(0.07717 \text{ kg/m}^3)(0.46 \text{ m})(5 \text{ m/s})}{9.359 \times 10^{-6} \text{ N/m}^2} \\ &= 18964.7 \end{aligned}$$

$$Nu_D = 76.53$$

$$\bar{h}_c = 31.97 \text{ W/m}^2\cdot\text{K}$$

Calculate the Biot Number:

$$Bi = \frac{\bar{h}_c L}{K_{\text{steel case}}}$$

Appendix F: (cont'd)

where  $L$  is a shape factor ( $\sqrt{A}$ )

Neglecting ends:

$$L = \frac{\pi r^2 l}{2\pi r l + 2\pi r^2} = \frac{r l}{2(l+r)} = 0.159 \text{ m}$$

$$Bi = \frac{(31.97 \text{ W/m}^2\text{K})(0.159 \text{ m})}{35 \text{ W/mK}}$$

For simplicity, Transient Conduction with Negligible Internal Resistance will be assumed. Since  $Bi > 0.1$ , an error of over 5% is expected.

For TC w/ NIR:

$$q(t) = \bar{h}_c A_s (T(t) - T_{\infty})$$

As a worst case  $q(t)$  will be assumed constant at 2437 W.

Therefore:

$$2437 \text{ W} = (31.97 \text{ W/m}^2\text{K})(1.84 \text{ m}^2)(T(t) - T_{\infty})$$

$$42.28 \text{ }^\circ\text{K} = T(t) - T_{\infty}$$

## Appendix F: (cont'd)

For motor design specs, the motor temperature,  $T(t)$ , is not to exceed  $45^\circ\text{C} = 318^\circ\text{K}$

So:

$$42.28^\circ\text{K} = 318^\circ\text{K} - T_{\infty}$$

$$T_{\infty} = 276^\circ\text{K} = 3.0^\circ\text{C}$$

Therefore  $T_{\infty}$  of the system must start at or below  $276^\circ\text{K}$  and never exceed this value to keep the motor from overheating.

The total system will be assumed to start out at  $T_{(0)} = T_{\infty} = 276^\circ\text{K}$ .

Going back and calculating the actual temperature of the motor with hydrogen at  $276^\circ\text{K}$ :

Properties of hydrogen at  $276^\circ\text{K}$ :

$$\alpha = 1.350 \text{ m}^2/\text{s}$$

$$k = 0.1696 \text{ W/mK}$$

$$\rho = 0.03969 \text{ kg/m}^3$$

$$\mu = 8.462 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

$$Pr = 0.709$$

$$C_p = 14191.6 \text{ J/kg}\cdot\text{K}$$

$$Re_D = \frac{\rho D V}{\mu} = 24378.0$$

$$Nu_D = 88.95$$

$$\bar{h}_c = 32.80 \text{ W/m}^2\cdot\text{K}$$

Appendix F: (cont'd)

$$Bi = \frac{\bar{h}_c L}{k_{\text{steel}}} = \frac{(32.8 \text{ W/m}^2\text{K})(0.159 \text{ m})}{35 \text{ W/mK}}$$
$$= 0.149$$

For TC w/NIR:

$$2478 \text{ W} = h_c A (T(H) - T_{\infty})$$

$$2478 \text{ W} = 32.8 \text{ W/m}^2\text{K} (1.84 \text{ m}^2) (318^\circ\text{K} - T_{\infty})$$

$$T_{\infty} = 277^\circ\text{K}$$

So, temperature rise of hydrogen as it flows over the motor is:

$$\Delta T = 277 - 276 = 1^\circ\text{K}$$

To ensure this before operation, thermocouples must be placed on the different components of the system.

In order to keep  $T_{\infty}$  at or below  $276^\circ\text{K}$ , heat must be removed from the hydrogen at the same rate that it is being added, i.e.  $2478 \text{ W}$ .

## Appendix F: (cont'd)

It is proposed to absorb heat from the hydrogen by a heat sink made with banks of aluminium cylinders and then radiate this heat into deep space.

For the banks of tubes to radiate  $2478 \text{ W}$ , at  $T = 276^\circ \text{K}$ , the area of the sink exposed to free space must be:

$$\begin{aligned} q_r &= \sigma A_s T^4 \\ 2478 &= (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4) A (275.7^\circ \text{K})^4 \\ A &= 7.67 \text{ m}^2 \end{aligned}$$

For cylinders arranged as in the figure it will be estimated that the fluid temperature has risen to  $280^\circ \text{K}$ .

Some Properties of hydrogen at  $280^\circ \text{K}$ :

$$\begin{aligned} \alpha &= 1.384 \times 10^{-4} \text{ m}^2/\text{s} & \mu &= 8.545 \times 10^{-6} \text{ N}\cdot\text{s/m}^2 \\ k &= 0.172 \text{ W/m}\cdot\text{K} & Pr &= 0.709 \\ \rho &= 0.08839 \text{ kg/m}^3 & C_p &= 14212 \text{ J/kg}\cdot\text{K} \end{aligned}$$

To compute the Reynold's Number, a velocity of  $10 \text{ m/s}$  is assumed.

$$Re_D = \frac{(0.15 \text{ m})(0.08839 \text{ kg/m}^3)(10 \text{ m/s})}{8.545 \times 10^{-6} \text{ N}\cdot\text{s/m}^2} = 15516$$

## Appendix F: (cont'd)

The average Nusselt Number for the tubes:

$$\overline{Nu}_D = \overline{Pr}^{.36} (.27 Re^{.63})$$

$$\frac{\overline{h}_c L}{k} = 104.2$$

$$\overline{h}_c = \frac{(0.172 \text{ W/m}^2\text{K})(104.2)}{0.15 \text{ m}} = 119.5 \text{ W/m}^2\text{K}$$

The Biot Number is:

$$Bi_c = \frac{\overline{h}_c L}{k_{alum}}$$

$$\text{where } L = \frac{r_o}{2(r+1)} = \frac{(0.075 \text{ m})(0.3 \text{ m})}{2(0.375 \text{ m})} = 0.03 \text{ m}$$

$$Bi_c = 0.022$$

since  $Bi_c \ll 1$ , TC w/NIR is assumed.

$$q(t) = \overline{h}_c A_s (T(t) - T_{\infty})$$

$$-2478 \text{ W} = (119.5 \text{ W/m}^2\text{K}) A_s (276^\circ\text{K} - 277^\circ\text{K})$$

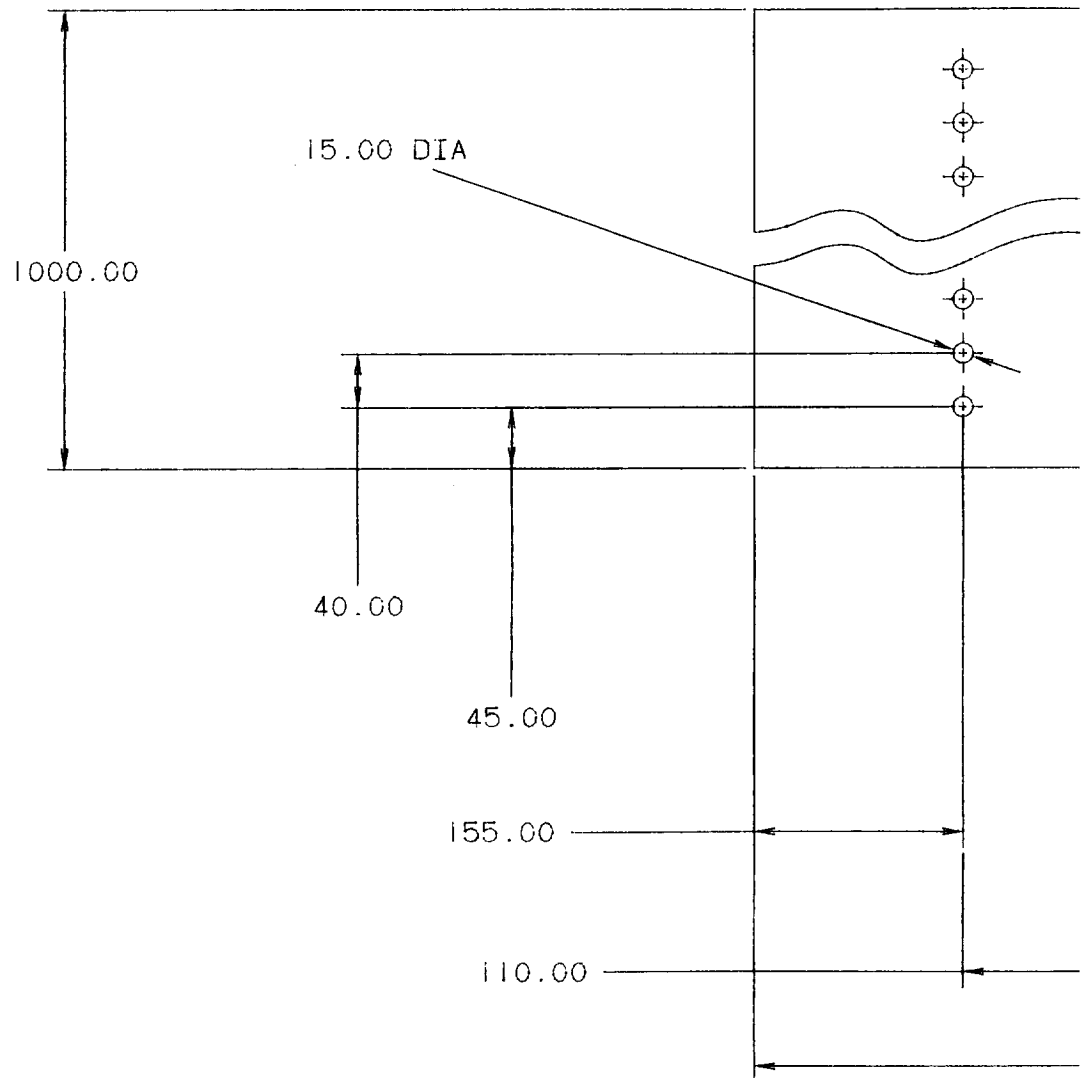
$$A_s = 20.7 \text{ m}^2$$

Since each cylinder has a surface area of:

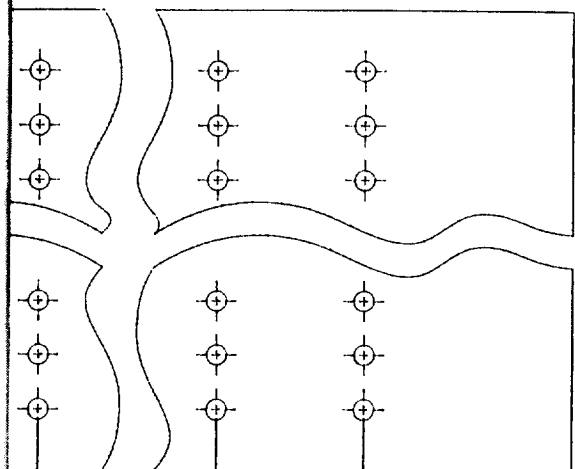
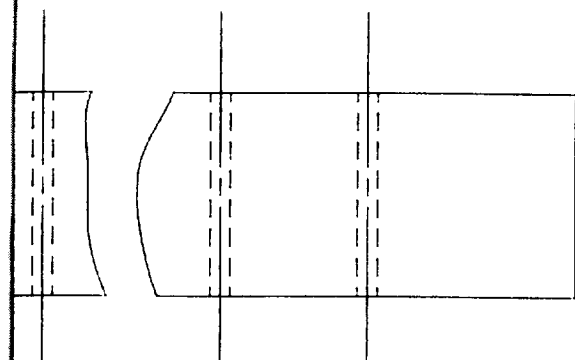
$$SA_{cyl} = \pi(0.15 \text{ m})(0.3 \text{ m}) = 0.141 \text{ m}^2$$

The number of cylinders is 147.

300.00



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155.00

110.00

8000.00

GEORGIA TECH	
COLLEGE OF ENGINEERING	
TITLE: RADIATOR	
DESIGN:	DATE 3/9/86
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## Group 2

### A Transportable 100-Meter Multiple-Application Lunar Drill

February 3, 1986

We have thus far divided our project into three sub-projects with two group members working on each sub-project.

The divisions are:

- 1) Drill bit, dust removal system, and internal cooling system.
- 2) Motor drive and rigging system.
- 3) External cooling system for drill bit and motor.

Each sub-group is currently working on detailed research for published data on their subject. The entire group is also maintaining constant communication to assure that each subsystem will sufficiently interface in our overall design.

We have completed or on-line data base search through the library. Also, we have thoroughly searched and documented sources currently available in Price Gilbert Library.

## Group 2

### A Transportable 100-Meter Multiple-Application Lunar Drill

February 10, 1986

The design of the 100-meter drill is progressing as follows: Each subgroup has broken down its subsystem into an outline of parts to be designed. The overall group has met and will continue to meet often to assure that each subsystem will combine to make the best total design.

Each subgroup is also continuing detailed research to support its specific designs. Pre-published reports and designs have been found to be helpful in choosing optimum materials and methods.

## Group 2

### A Transportable 100-Meter Multiple-Application Lunar Drill

February 17, 1986

The drill bit and auger system has been designed and tentatively dimensioned. A replaceable diamond/matrix cutting system will be used. The replaceable section will be threaded into the main shaft and will be secured with set screws. An auger system for quick chip removal will be implemented. The system will be efficient enough to eliminate the need for internal bit cooling.

The bit removal system will be housed inside the main shaft. This system will be a collet arrangement which will grip the core each time the shaft is withdrawn from the hole.

The drill rig anchor system will consist of three expansion anchor mechanisms. One of these will anchor each leg and must hold up to 667 pounds. They are designed to hold 1000 pounds.

The motor cooling system will be a fluid circulation and radiator scheme. Since a hermetic motor design will be implemented, the cooling fluid will be in direct contact with the electric motor coils. The fluid will then pass through a radiator facing deep space to radiate the heat removed from the motor.

The motor itself will be some type of electric motor which can deliver up to 30 hp at 700 rpm. The motor will be hermetically sealed for ease of cooling and to protect it from harmful radiation.

The downward thrust mechanism is still under discussion. The possible solutions include variable pitch rollers to create downward thrust, or a hydraulic linear actuator, or a cable system. The thrust mechanism will deliver between 400 and 2000 pounds downward and will also be used to remove the shaft and core from the hole.

## Group 2

### A Transportable 100-Meter Multiple Application Lunar Drill

February 24, 1986

The dimensions of the bit section and shaft sections have been finalized. After consulting David Federer, President of Data Foundation Systems, Inc., it has been decided to use a matrix bit with 24 thin teeth. This scheme will allow sufficient chip removal as well as the advantage of a self-sharpening bit. Also, the material for the shaft has been selected on a strength:weight ratio basis. The material will be 2024-T4 aluminum. The dimensions were selected using design criteria which varied with various design parameters. The final set of dimensions was chosen by a minimum weight criteria.

The drill rig anchoring system has been finalized. It will consist of three auger anchors, one for each leg of the rig. Each auger will be 1.5 meters long and will hold up to 1300 pounds of pull.

The electric motor cooling system will consist of a reserve tank of nitrogen which will feed into the motor environment as heated gas is purged from the environment. The purge valve will be temperature controlled to release heated gas and will close once cool gas has lowered the motor temperature.

The specific motor choice has not been finalized but the motor performance requirements have been determined.

The drill rig frame will probably be a standard frame with a proven performance record. The frame may be altered slightly to conform to the specifications of this project.

## Group 2

### A Transportable 100-Meter Multiple Application Lunar Drill

March 3, 1986

All the details of the project have been addressed. Each subgroup is finishing calculations, drawings, and writing.

The final bit chosen is a self-sharpening diamond/matrix bit. A few modifications of the weight and dimensions of the auger sections and drill string have been made.

The drill rig anchoring system will remain an auger system for each leg of the rig. The anchor dimensions have been finalized.

The final motors have been selected. They are all General Electric motors with specifications to fit the parameters of the project. A scheme of three motors has been chosen. One motor will rotate the shaft, another will apply downward force on the shaft, and the third will raise the drill string after each two-meter section has been cut.

The drill rig will be an aluminum tripod. It will be anchored at each leg and have a vertical stroke slightly over four meters.

The motor cooling system will be a fluid circulation system. Nitrogen will be circulated through the motor housings and then through a heat exchanger which contains several aluminum rods which act as heat sinks. Each container of rods will hold a certain amount of heat before reaching a predetermined critical temperature. Once this temperature is reached, a fresh container of cool rods will replace the heated ones. The heated ones must then be transported to a cooling container, probably in the lunar base. The heat sink containers are reusable and offer the best solution for the heat transfer problems associated with this project.

## Group 2

### A Transportable 100-Meter Multiple-Application Lunar Drill

March 10, 1986

The project has been completed. As many drawings as possible have been done on the CAD system. Calculations on each section of the project have been formalized and included in the appendices of the technical paper.

The drill bit is a diamond/matrix bit bonded to a metal base. The auger shaft and drill string are aluminum. The core retrieval system is included in the drill string.

General Electric motors have been selected to drive the drill string. A tripod drill rig anchored at each leg will be used. A cuff to secure the drill string during its removal from the hole and its replacement in the hole will be used at the top of the hole.

A hydrogen-circulation system will be used to cool the motors. A hydrogen will be cooled using a radiator. This method of cooling is thought to be the most sophisticated although several others methods have been investigated.

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References (cont'd)

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